Power and The Engineer

Vol. 32 NEW YORK, OCTOBER 4, 1910 No. 40

SEE the chief!

Is that the chief, he in the faded, soiled blue jeans and jumper with the crumby old peak cap on his head?

Yes, that is the chief. He has just finished eating his lunch in that corner saloon where they give the largest "scoop" in the city for five cents; where they put up a free hot lunch all day; where sawdust saturated with tobacco juice and stale beer sends up a pungent aroma which mingles with the nutritious whiffs from the "hot-dog" cauldron and the smoking platter of freshly fried crabs; where the multitudes of flies act just as though they were regular cash customers, and where the conversation is frank and profanely emphatic even if it is not beneficial.

See the manly (?) swagger of the chief's walk. Fine dignity! The chief seems to show by his manner that he is proud of the fact that he is not dressed as well as the poorest paid clerk in the office upstairs.

First impressions are the most lasting and hardest to change. A man's personal appearance—his dress and manner—is invariably the basis of the first impression which he gives to other men. If a man looks slovenly and careless, other men will accept him as such until he proves himself to be otherwise. If he has a clean-cut, snappy appearance, he is judged to be straightforward, alert and onto his job until greater intimacy serves to reverse this decision. From your own experience you all know this to be so. How often, after turning a man down, have you not remarked "I didn't like his looks"?

Of course we know that the engineer is a busy, important man, and that his work often necessitates his getting a bit greasy and mussed up. Many reason in this way, "If I clean up I'll only get dirty again; few people see me, and, anyhow, what's the use!"

If we eat, we only get hungry again. But, whether a man eats or not, as far as he is concerned, things will be all the same a thousand years from now, so what's the use! Never heard anyone reason that way, did you?

Some of the most trivial incidents often have far-reaching effects. It is said that the simple circumstance of a spider weaving its web across the opening of a cave into which a Scotch patriot had just crawled to hide saved the patriot's neck, and, in all probability, the cause for which he was fighting. The cave didn't "look" as though Bruce were hiding there and so the trackers passed it by.

Because he did not "look" as though he had some *quality* hiding within, many an engineer has been passed by when the "trackers" after brains came his way.

Why handicap yourself by manitaining an unprepossessing appearance to be lived down, when a favorable impression may be given for very little effort and at a very small cost?

An engineer doesn't have to wear a prince albert, mauve spats and gloves and a "high lid" to convey the impression that he is wide-a-wake and onto-his-job. In fact, one who showed up to work in such regalia would undoubtedly be transferred to the "boo-boo" house in short order. We will not give a list of "what to wear" or instructions on "how to trim the beard." We don't think it necessary, anyhow. There is lots more that could be said along this same line but don't wait—get busy and,

LOOK YOUR PART.

Plant of the Bigelow Carpet Company

By B. S. Walters

A new central station plant furnishing power for the two mills of the Bigelow Carpet Company which are located at some distance apart, and each of which was formerly operated by its individual power plant at a considerable sacrifice in economy.

The Bigelow Carpet Company, of Clinton, Mass., consists of two separate mills; one located on the Worcester & Nashua division of the Boston & Maine railroad and known as the "spinning mill," where wool scouring, spinning and the preparation of the yarn is accomplished; the other mill, known as the "Axminster weaving mill," where the dyeing and hacking of the yarns and the weaving of rugs and carpets is carried on, is an eighth of a mile distant from the spinning mill. Until recently each mill contained its individual power plant, and all current for lighting was furnished by the weaving mill. The company, after careful investigation by their engineer, Charles T. Main, of Boston, decided that considerable economy in operation could be secured by concentrating the power-generating units into one plant; consequently, plans were immediately drawn and a new power plant erected. A further consideration concerning the erection of a central power plant was the anticipated extension of the weaving mill.

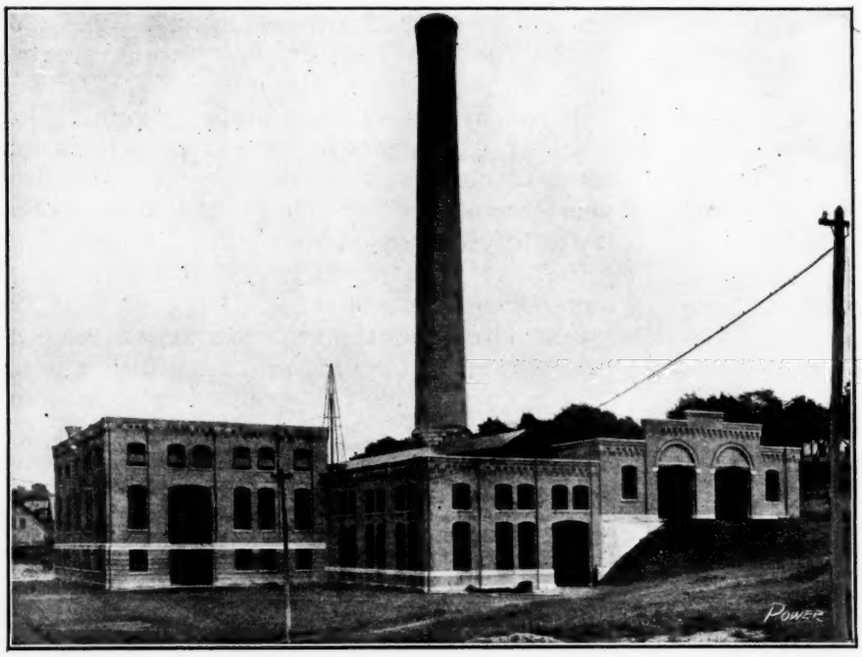


FIG. 1. EXTERIOR VIEW OF PLANT

The new plant, an exterior view of which is shown in Fig. 1, is located in a lot about halfway between the two mills, where advantage was taken of a slope of the ground for the arrangement of a coal pocket. A spur track from the railroad was laid on the top of the embankment, and a trestle was extended from this into the power house. The tracks enter the coal pocket 18 feet above the level of the boiler-room floor, and the coal is dumped from the cars to the bunker floor, where it is loaded by hand into industrial cars and run through doors in the 20-inch fire wall which separates the coal pocket from the boiler room. Each opening is equipped with a vertical rolling steel door and in front of each opening is a pair of two-ton scales with the registering device inclosed in a dust-proof case.

The coal pocket is built of reinforced concrete with structural-steel columns, brick walls and steel trusses to support the roof. It is 71x197 feet, and has a maximum capacity of about 5000 tons. It is divided in the middle by a reinforced-concrete fire wall extending to the level of the tracks.

The chimney, which is located in the coal pocket on the side next to the boiler room, is built of red brick, has a 10-foot flue, and rises 200 feet above the boiler-room floor. The boiler room, which adjoins the coal pocket, has a floor area 50x197 feet and a clear hight of 23 feet 6 inches to the bottom of the trusses. The boilers face the coal pocket, there being a firing space 20 feet deep between the boiler front and the dividing wall. All the firing is done direct from the industrial cars, no coal being dumped upon the boiler-room floor, also all the coal is accurately weighed upon being brought from the coal pockets.

The boiler equipment consists of eight old boilers, which were taken from the Axminster mill plant, and six new boilers with space available for the future installation of two additional boilers. The boilers are of the horizontal, return-tubular type, 84 inches in diameter, with 180 three-inch tubes 19 feet long, and all built by the D. M. Dillon Water Works, of Fitchburg, Mass. Fig. 2 is an end view of one of the boilers showing the piping at the back of the battery.

The engine-room equipment consists of a 26x48x42-inch Rice & Sargent, horizontal cross-compound condensing engine running at 120 revolutions per minute, which is direct connected to a 750-kilowatt Allis-Chalmers generator; a 20x36-inch Rice & Sargent simple noncondensing engine, running at 150 revolutions per minute and direct connected to a 300-kilowatt Allis-Chalmers generator; a 12x12-inch simple noncondensing Shepard engine, direct connected to a 50-kilowatt Allis-Chalmers generator; each of the foregoing generators being of the three-phase, 60-cycle, 600-volt type. A 12x12-inch simple noncondensing Shepard engine is direct connected to a 50-kilowatt, 120-volt direct-current generator, furnishing excitation current to the main generators. There is also a 50-kilowatt motor-generator set for excitation purposes. Space is provided in the engine room for an additional unit of 750 kilowatts capacity. All the engines

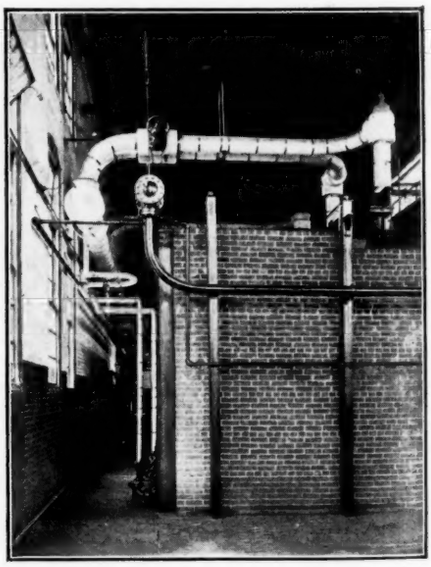


FIG. 2. PIPING AT THE BACK OF BOILERS

are supplied with oil by a "White Star" gravity oiling system.

A general view of the engine room is shown in Fig. 3.

There are two steam pressures used at the plant: a high-pressure system of 125 pounds gage providing steam for the main engines and auxiliaries, also supplying steam to the dye house; and a low-pressure system of from 5 to 10 pounds gage, supplying steam to the feed-water heater and the heating system of one mill. There is also a vacuum system in which the vacuum is maintained at the proper point to insure the desired temperature of the overflow water going to the dye house.

The pump-room equipment consists of two fire pumps, one of 1500 and the other of 1000 gallons capacity, each made by the International Steam Pump Company; three 14x8x12-inch duplex boiler-feed pumps for regular service and one 8x5x12-inch vertical piston duplex pump for night and Sunday service, all manufactured by the Warren Steam Pump Company, and an 8x7x12-inch duplex piston pump for supplying cold water to the plant. The boiler-feed and fire pumps are equipped with Locke governors and the service pump with a Fulton tank governor. A 3000-horsepower Kelley-Berryman feed-water heater is also located in the pump room. The boilers are provided with Williams feed-water regulators, which are a necessary appliance in this plant, owing to the irregular demands for steam in the dye house. All the hot water supplied to the boilers is measured by Hersey water meters, there being one meter for each battery of boilers.

As the supply of water is limited, its economical use for condensing and manufacturing purposes was very carefully considered when designing the plant. The suction of the boiler-feed pumps is connected to the overflow line from the condenser, the water in which is heated to about 95 degrees Fahrenheit. This line also connects with the dye house and has an overflow through an elevated loop through which any excess water is discharged into the pond from which it is drawn to the condenser.

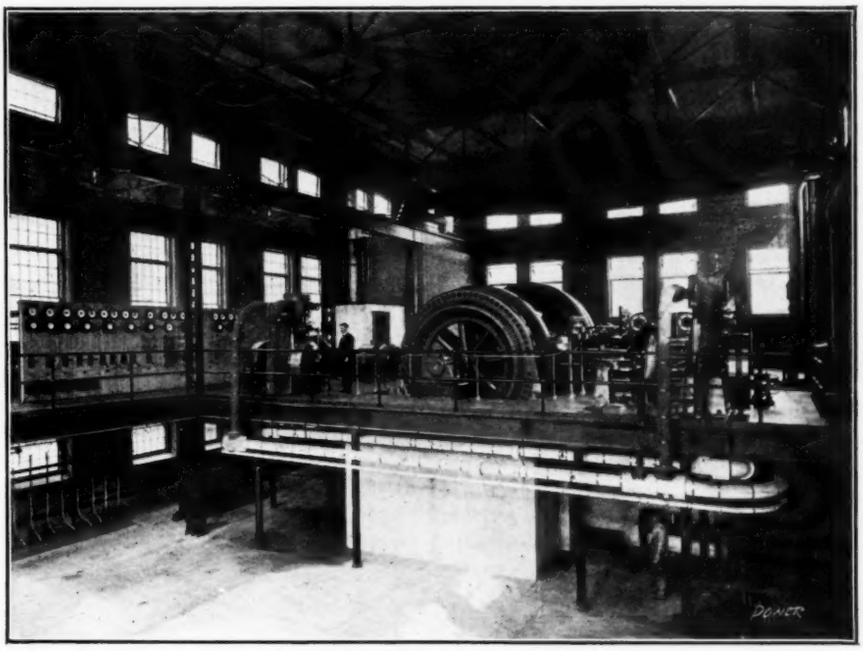


FIG. 3. ENGINE ROOM, INCLUDING SWITCHBOARD GALLERY

A test which was recently conducted on two of the boilers showed the following results:

BOILER TEST ON TWO 84-INCH DILLON BOILERS.

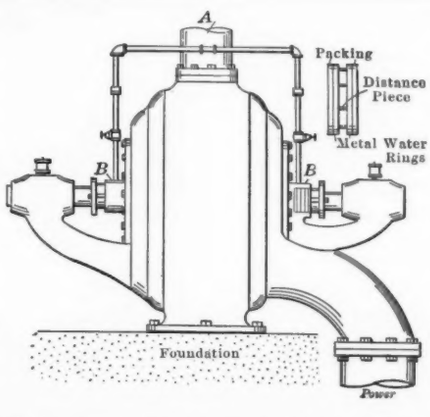
Duration of test	10 hours, 20 minutes
Method of conducting test	Alternate
Kind of furnace	Brick, with Dillon shaking grates
Grate surface, square feet, 2 × 46.7	93.4
Water heating surface, square feet, 2 × 2,741	5,482
Ratio of heating to grate surface	58.7 : 1
Heat value of dry coal per pound, B.t.u.	14,829
Moisture in coal, per cent	1.28
Ash and refuse in dry coal, per cent.	6.85
Dry coal consumed per hour, pounds	2,235
Dry coal per square foot of grate surface per hour, pounds	23.9
Water evaporated per hour, corrected for quality of steam, pounds	20,470
Factor of evaporation	1.1657
Equivalent evaporation per hour, from and at 212 degrees Fahrenheit, pounds	23,860
Equivalent evaporation per hour from and at 212 degrees Fahrenheit per square foot of heating surface, pounds	4.35
Steam pressure, gage	109.1
Temperature of feed water entering boiler, degrees Fahrenheit	93.2
Temperature of escaping gases, degrees Fahrenheit	388
Moisture in steam, per cent	2.66
Boiler horsepower developed	715
Water apparently evaporated under actual conditions per pound of coal as fired, pounds	9.28
Equivalent evaporation from and at 212 degrees Fahrenheit per pound of coal as fired, pounds	10.52
Equivalent evaporation from and at 212 degrees Fahrenheit per pound of combustible, pounds	11.45
Efficiency of boiler and grate, based on dry coal, per cent	69.5

Continuous readings were taken on a CO_2 recording machine and showed the CO_2 to range from 6 to 15 per cent., with an average of 12 per cent. for the test.

Water Seal for Centrifugal Pump

Centrifugal pumps cause more or less trouble by drawing in air through the stuffing box around the shaft. Different methods of water sealing have been tried by engineers with varying success. The method adopted by the Chattanooga (Tenn.) Power and Light Company, which is illustrated in the accompanying drawing, has given satisfaction.

The outlet pipe is tapped at *A* and from this point a branch pipe is run to the bearing on either side, as shown at *B*. In place of the regulation method of packing the stuffing box a combination is used, consisting of one ring of ordinary packing placed next to the inner side of the stuffing box, followed by two rings of brass each 1¾ inches thick and spaced by distant pieces 1 inch apart. On the outside of this combination of packing rings another piece of ordinary ring packing is fitted next to the stuffing-box gland. Water from the small pipe flows into the space left between the two metal rings, thus producing a water seal around the shaft.



WATER SEALING A CENTRIFUGAL PUMP

In the case cited the engineers do not recommend that the stuffing box be screwed up so tight that no leakage will occur; on the other hand, they prefer that a slight leakage take place so that the soft packing will not deteriorate too rapidly. The idea works out well in the present instance.

The Predecessors of Watt's Engine

By H. W. Dickinson

Salient points in the history of the atmospheric engine of Thomas Newcomen, which did good work for a period of more than fifty years.

Most educated persons and probably very many engineers would, if asked who James Watt was, reply at once that he was the inventor of the steam engine, and while this is in a sense the truth, it is not the whole truth, because Watt commenced as the improver of an existing apparatus—to wit, the atmospheric engine of Thomas Newcomen.

In this article I propose to give the salient points in the history of this remarkable engine, which did yeoman service for more than 50 years, and leave the subject at the stage when Watt's attention was drawn to it and when he began his improvements.

The genesis of an invention is always interesting, but of this, the most momentous discovery of its age and the ultimate source of the material progress of the last two centuries, it must be regretfully owned that very little is known. All that can with certainty be said is that in 1719 or 1712 the Newcomen engine was in actual operation.

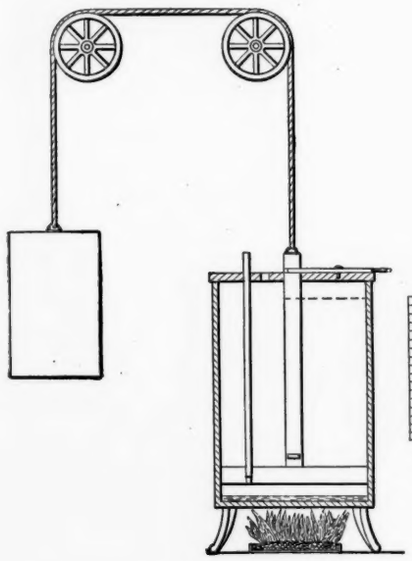


FIG. 1. DENIS PAPIN'S ENGINE—1690

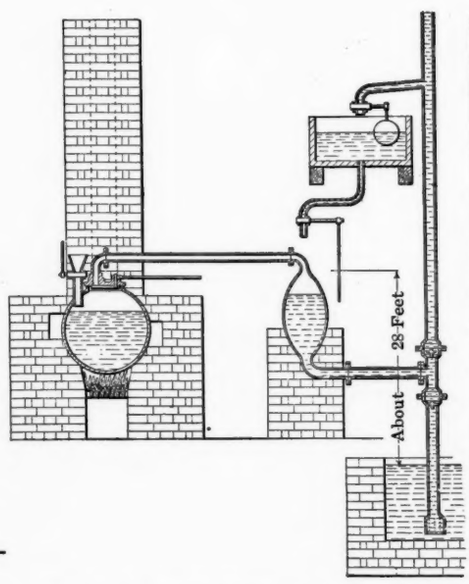


FIG. 2. THOMAS SAVERY'S ENGINE—1698

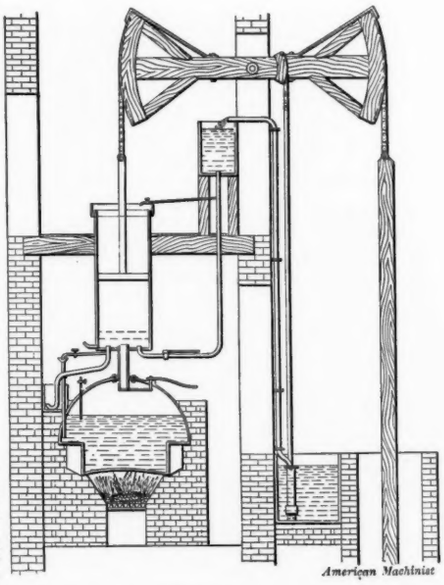


FIG. 3. NEWCOMEN'S ENGINE—1712

THE WORK OF PAPIN AND SAVERY

For quite 50 years, scientific and practical men, notable among them Denis Papin and Thomas Savery, had been engaged in trying to turn to practical account the discovery of Evangelista Torricelli that the atmosphere had weight, a fact that he had deduced from his observations of the hight that mercury will stand in a barometer tube. Then was given for the first time the true reason why water follows the bucket of a suction pump. It was not a great step from this to imagine that if one could remove the pressure of the atmosphere from under the bucket, the pressure of the air on the upper side would force the bucket down and, by attaching a cord to the bucket, lift a weight.

Papin got so far in 1690 as to scheme a closed vessel or cylinder with a piston, in which was a nonreturn valve, as shown in Fig. 1. A little water on the bottom of the cylinder was boiled and when the steam had forced out the air through this valve, and when the cylinder had cooled again, the piston would descend when the catch was released. But he was unable to devise means for repeating the motion even had it not been altogether too slow.

Savery went on another track. His apparatus had a boiler with a hand-controlled valve admitting steam to a receiver provided with foot and head valves. By external condensation of the steam by a stream of water outside, he could draw water from a depth of about 20 feet, and then shutting off the condensing water force it to a hight depending on the steam pressure, all as shown in Fig. 2. The necessity for this pressure was a fatal defect at this stage of the mechanic arts, for no means of construction which would stand more than a very moderate pressure were known; it had to be a plumber's job, in fact. However, Savery managed in 1698 to get a patent for 14 years on his engine; indeed, it was a master patent, so widely was it worded: that is, "for raising of water and occasioning motion to all sort of millworks by the impellent force of fire." He was granted a prolongation of his patent for 21 years, making 35 in all, so that it did not expire till 1733. The invention, for the reason given, met with but scant success. The curious in these matters will notice that this apparatus is practically identical in principle with that brought out in 1872 by Cyrus H. Hall, and known as the pulsometer.

THE NEWCOMEN ENGINE

And now Newcomen comes on the scene, for we are assured by a contemporary writer that: "Mr. Newcomen was as early in his invention as Mr. Savery was in his," but the latter being nearer to the court obtained a patent first. Newcomen's invention was totally different from Savery's, yet the latter's claim fully covered it, and this accounts for the fact that Newcomen never did get a patent, repeated statements to the contrary notwithstanding. The writer already quoted says that Newcomen was glad to come in as a partner of Savery. For about 12 years, therefore, Newcomen, who was an ironmonger, assisted by John Cawley or Calley, a glazier, was perfecting the invention at his home in Dartmouth, Devon.

In its essentials the plant, shown in Fig. 3, consisted of a boiler, after the style of a brewing copper, above

which was a thin brass cylinder provided with a piston attached by a chain to one end of a beam or gigantic pump handle, from the other end of which hung the buckets and pump rods that balanced the piston. Steam just above atmospheric pressure was introduced into the cylinder, and when the piston had got nearly to the top of its stroke a jet of water was turned on inside the cylinder, the steam was condensed and the atmosphere forced down the piston, thus raising the pump bucket. The grand feature of this invention was that no matter from what depth water was to be raised, steam at atmospheric pressure only was required.

So far this was a combination of known elements, but success would not have resulted from it without two other beautiful and original inventions. The first was a valve gear which opened and shut the steam and injection valves at the right moments; the second was a "snifting" valve whereby the injection water and the condensed steam, together with the small amount of entrained air, were removed from the cylinder. This valve gear was the first invention in the world's history by which a nonliving agent had been schemed to repeat its own operations automatically an indefinite number of times—in its results surely a most pregnant discovery. Nevertheless, it seems to have excited the attention of the scientific men of the day very little.

From contemporary prints and published notices we are able to piece together with some show of probability the subsequent career of this invention. The earliest known representation of the engine is an engraving, dated 1719, of an engine erected in 1712. It is not quite certain, however, that the engine shown was the first to be erected, because proposals were made in 1710-1711 to put down an engine at Griff, in Warwickshire. Even if not at once acted upon, they were soon after, because an engine was in existence at Griff a few years later. Difficulties were encountered in the construction and working of these pioneer engines, but these were overcome by skilled mechanics from Birmingham.

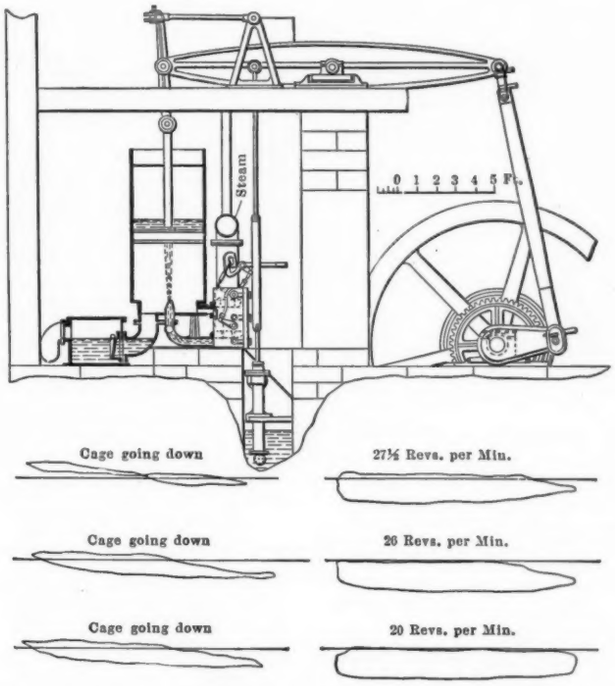


FIG. 5. ELEVATION OF NEWCOMEN ENGINE SHOWN IN FIG. 4 WITH INDICATOR CARDS TAKEN JUNE 20, 1901

FIG. 4. A NEWCOMEN ENGINE THAT IS STILL DOING DUTY

THE HUMPHREY POTTER STORY A MYTH

It is said that Newcomen at first condensed like Savery, by water on the outside, and that the jet was only found out by the accident of a leaky piston. To understand this it should be remarked that water was used on the top of the piston as packing—another ingenious idea.

We are assured by Dr. J. T. Desaguliers in his "Experimental Philosophy" that the self-acting valve gear was schemed by a boy named Humphrey Potter because he wanted to be released from turning the cocks by hand so he could go and play—a most improbable tale. Several individuals of the name of Potter appear in connection with the Newcomen engine during the subsequent 15 years of its existence; indeed one went as far afield as Königsberg, in Hungary, to erect an engine, and was there believed to be the inventor of it. No doubt one of these engineers added an improvement, and hence the story.

The snifting valve was a neat way of overcoming "wind logging." From the bottom of the cylinder an eduction pipe led to a cistern of water where the end was closed by a poppet valve. When steam entered the cylinder the condensed water ran out by gravity and the air followed through this valve, giving rise to the noise which gave it its name.

SIZE AND COST OF THE OLD ENGINES

A rough idea may be formed of the space necessary at this period to develop a given power when it is noticed that the

engine shown in the above mentioned engraving had a cylinder 21 inches in diameter by 7 feet stroke and from the quantity of water raised developed 5½ effective horsepower. The cost, too, was enormous. For an engine not much larger than this, built in 1725 in Scotland, the cylinder alone cost £250, and the total expense, excluding the engine house and pit work, exceeded £1000. As this must be multiplied by 3 or 4 to obtain the value at the present day, it will be seen that it was no light undertaking to erect an engine in the year of grace 1725. The engine met a long-felt want, so that, notwithstanding all this, it was adopted at colleries up and down the country, to such an extent that when the celebrated engineer John Smeaton turned his attention to the subject he found, in 1769, no less than 57 engines, one with a cylinder as large as 75 inches diameter, in use in the Newcastle district alone. The average duty of these engines was 5.6 millions of pounds of water raised 1 foot high by the expenditure of one bushel (84 pounds) of coal. By judicious proportioning and better workmanship Smeaton obtained a duty of 9½ millions. The most celebrated engine of his construction was that at Chacewater in Cornwall, completed in 1775. By that time, however, Watt, in partnership with Boulton, had begun the construction of his improved engine and as it would "fork" as much water as the atmospheric engine with about one-third of the consumption of fuel, the doom of the latter was sealed.

It did not go under without a struggle for, at the pit mouth where coal is cheap, these engines long survived. Indeed one is actually at work at the present day at a colliery near Rutherglen in Scotland. It is shown in Figs. 4 and 5, for the latter of which I am indebted to the *Proceedings* of the Institution of Mechanical Engineers. It is a rotative engine and from the illustrations it will be noticed that the wooden beam has been replaced by one of iron, and the flexible connection to the piston by a piston rod and parallel motion. As one would expect, the beam was the weakest part of the engine.

The long life of these veterans is in marked contrast to the engines of today, which find their way to the scrap heap before they have had a chance to become venerable.

Types of Ammonia Compressor

By F. E. Matthews

Of the two types of ammonia compressor, double- and single-acting, the former is usually a horizontal and the latter a vertical machine. Typical examples of each type are presented.

Ammonia compressors are divided into two principal classes; double-acting and single-acting. The former type is most commonly horizontal, although frequently of vertical construction. The single-acting type is almost exclusively a vertical machine. Each type has its own followers among builders, and under certain conditions possesses some advantages over the other. While there is much variation in details of design among the various builders, the accompanying illustrations, Figs. 1 and 2, are the most characteristic of the general types. Fig. 4 shows a modification of the vertical single-acting machine which may be said to be typical of the compressor of small capacity.

Vertical Single-acting Compressors

The accompanying illustration, Fig. 1, giving a lateral elevation in section of a characteristic vertical single-acting ammonia compressor, shows the relative arrangement of compressor and engine cylinders as well as the principal details of design. In this type of compressor the vaporized refrigerant enters the compressor near the bottom, passes up through the suction valve, located in the compressor piston, during the downstroke, and is compressed and discharged through the discharge valve, located in the safety head, during the upstroke of the piston. The compressor may be water jacketed or not; popular preference, however, is in favor of the water jacket, and most machines are so built.

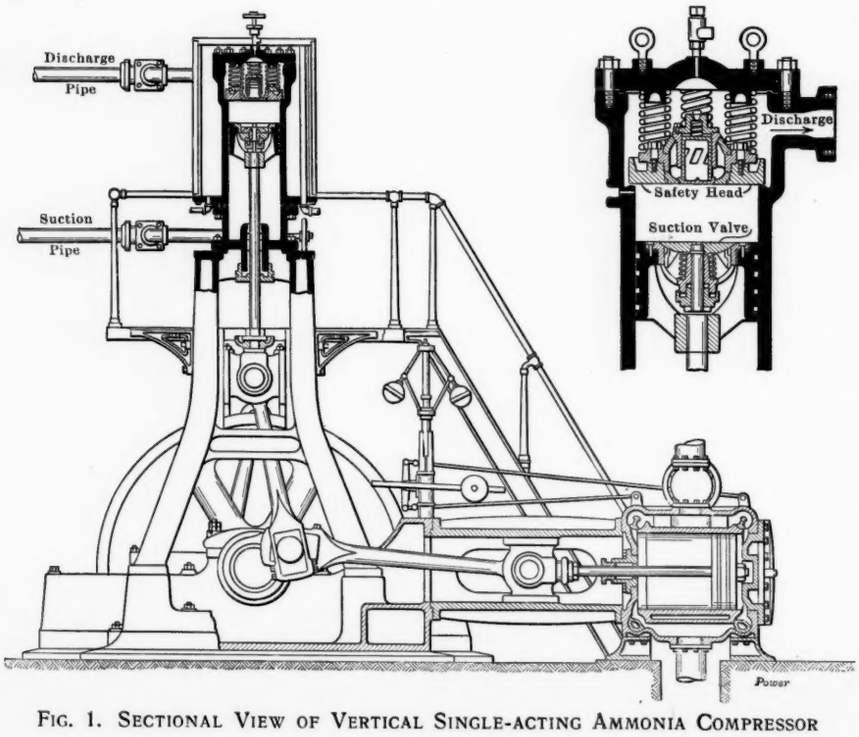


Fig. 1. Sectional View of Vertical Single-acting Ammonia Compressor

Vertical compressors possess the advantage of requiring less floor space than horizontal machines and the disadvantage of being less accessible for repairs. The inaccessibility of suction valves located in compressor pistons is offset by the unmistakable advantage offered by this type of machine in that these suction valves can be made of generous area, and the inertia of the valve tends to hasten its closing promptly as the piston reverses at the lower end of its stroke, thus preventing opportunity for gas to escape from the cylinder during the time required for the acting of stationary valves (unless they be heavily spring loaded, which tends to prevent the back pressure in the cylinder from quite reaching the hight of that in the suction line from the coolers). Inertia also tends to open the valve immediately when the piston begins its downward stroke, giving full opportunity for the cylinder to fill. The spring below the suction valve should be of such strength as to almost balance the weight of the valve, so that its inertia may act promptly at each end of the stroke.

Vertical single-acting compressors are usually provided with a "safety head" which is normally held securely to its

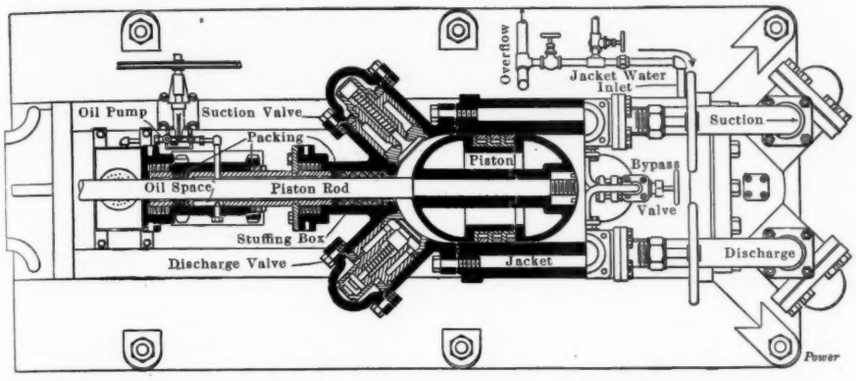


FIG. 2. PARTIAL SECTION OF HORIZONTAL DOUBLE-ACTING AMMONIA COMPRESSOR

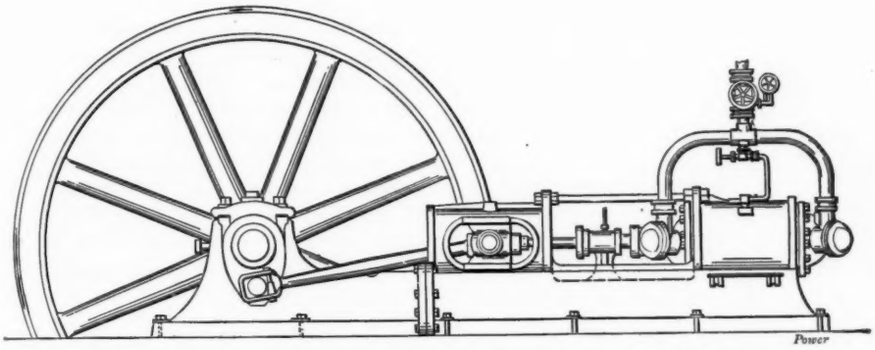


FIG. 3. TYPICAL HORIZONTAL DOUBLE-ACTING AMMONIA COMPRESSOR

seat by strong springs, but which, in the event of abnormal quantities of liquid ammonia or broken parts entering the cylinder above the piston, are pushed back, compressing the springs and thereby saving the machine from the strains that would otherwise occur. One of the principal advantages claimed by the advocates of the single-acting compressor is that the use of the safety head allows the compressor pistons to be operated with less clearance than would be practicable in the case of double-acting machines, a condition which insures a more complete expulsion of the gas.

THE HORIZONTAL DOUBLE-ACTING MACHINE

Fig. 2 represents a horizontal half section of a characteristic horizontal double-acting ammonia compressor. The right-hand portion of the cut shows the exterior of the head end of the compressor-cylinder valve housings, suction and discharge connections and valves. The remaining portion of the cut shows the details of construction of the compressor cylinder, water jacket, piston suction and discharge valves, double stuffing box and means of lubricating the piston rod. The outer wall of the water jacket is formed by the main frame casting, which is bored and fitted with a working cylinder liner consisting of a straight sleeve forced into place by hydraulic pressure and bored to the required size. The valves in this type of compressor are arranged radially to the hemispherical cylinder heads. The piston rod is provided with a primary stuffing box where it enters the compression cylinder. The packing in this box is tightened by a primary packing nut which carries a long sleeve, the other end of which is provided with a secondary stuffing box and packing nut. The main stuffing box, containing the bulk of the packing, withstands the high pressure of the ammonia in the compressor cylinder. The small stuffing box at the end of the sleeve is provided with sufficient packing to withstand the pressure of the oil circulated by the oil pump through the hollow sleeve surrounding the piston rod, in order to insure constant lubrication of and to maintain an oil seal on the main stuffing box.

The general appearance of the horizontal double-acting compressor cylinders just described is shown in the longitudinal elevation, Fig. 3.

INCLOSED CRANK-CASE COMPRESSORS

In addition to the two principal types of compressor previously described, the inclosed-crank type is deserving of mention because of the great number of such machines of small capacity now being installed. Details of design of this type of compressor are even more varied than those of the machines already described, and it is difficult to point out a single design that can be said to be more characteristic of the type than another.

In the illustration, Fig. 4, the refrigerant vapor enters the compressor cylinders through suction valves located in the cylinder head. Valves so located cannot be made of so liberal dimensions as

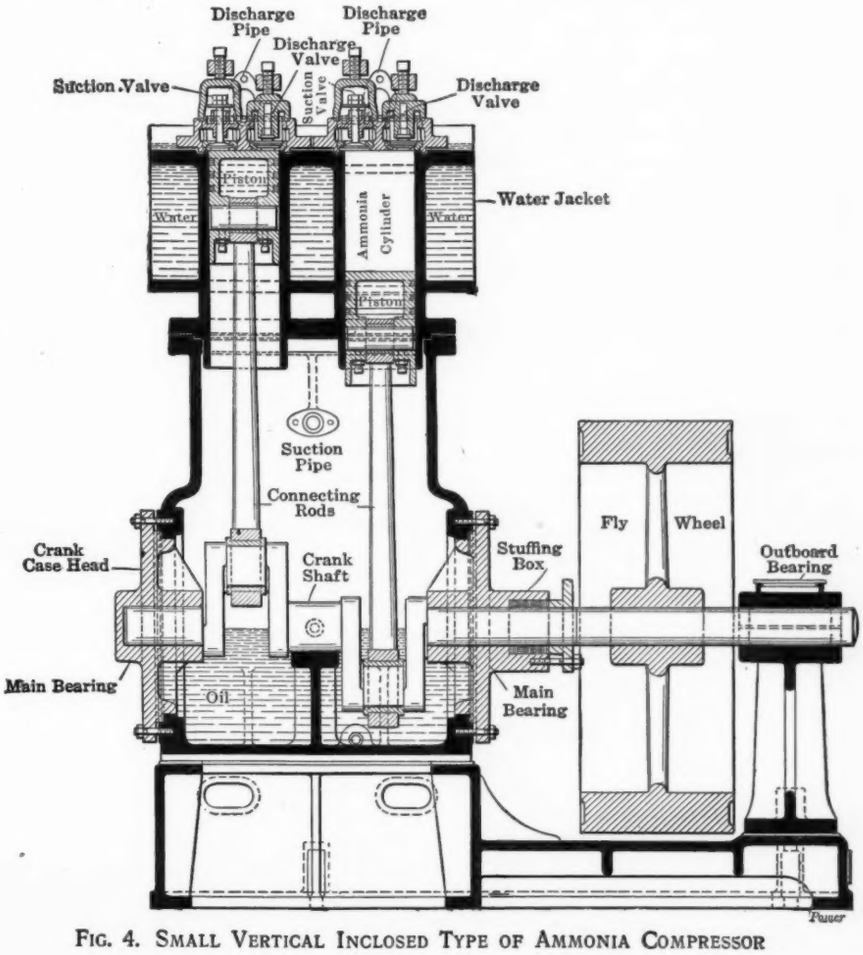


FIG. 4. SMALL VERTICAL INCLOSED TYPE OF AMMONIA COMPRESSOR

those located in the compressor piston, and the assistance which inertia offers in the way of opening and closing suction valves located in the piston cannot be realized. To offset this disadvantage, oil from the crank case is much less likely to be carried over into the condenser and low-pressure side of the system.

Machines of the inclosed type are especially adapted to use where little, or only inefficient, attendance is available. The details of construction which make less attention possible in this type of machine are principally the stuffing box and the main-bearing lubrication. The crank case being filled with oil to the center of the crank shaft, and the outboard bearing being usually ring oiling or provided with a compression grease cup, little attention to lubrication is necessary. There are no reciprocating piston rods to pack, the only stuffing box required being on the crank shaft, where it is always well lubricated and not subject to such extremes of temperature as are the pistons in other types of machine. The cylinder, however, must be cooled, and the water jackets for this purpose are plainly shown in Fig. 4.

The Theory of the Cooling Tower

By Joseph H. Hart

The author shows how it is possible for water to leave the tower at a temperature of 25 to 30 degrees below that of the air and explains how the cooling effect is obtained by absorption, conduction, convection and radiation.

Cooling towers operate as the result of two separate and distinct conditions, which make toward their cooling effect, and these are very often in opposition. Cooling towers are used to-day solely for the purpose of cooling water, which has become heated by passage through either a steam or ammonia condenser and is hence unfit for further use in this device unless it is itself in turn cooled off, in which case, however, it can immediately be used over again in the condenser with full effect. This, then, covers the entire scope of the cooling tower, to effect a saving in the water consumption, to the extent to which this is influenced by temperature variations in the water itself in both the steam and ammonia condensers. Of course, there are a number of small additional applications in industrial developments, in which similar conditions hold, where cold water is used for cleansing or purifying purposes, or even for the condensing feature in a number of other applications, but all of these are comparatively unimportant, and the chief use of the cooling tower is in condenser work in steam-power and refrigeration plants.

Here, the saving is confined to a diminution in the water consumption in the plant, and to a saving under some circumstances in the power consumption as well. In extreme cases where the water is extremely bad for feed-water purposes, cooling towers have been installed with a total increased power consumption as the result of their use, but with a greatly increased efficiency when all the determining factors have been considered. Again, cooling towers can be installed with considerable satisfaction with merely a power saving in view, if the pumping duty is large and the water circulation very unusual. This is quite possible, since there is invariably a saving in water consumption and consequently a saving in power on this account, as well as in the performance of the actual pumping itself. In a general way it can be said that wherever the water is bad or difficult to obtain, it pays to consider the possible installation of a cooling tower. Again, the question of mere size may enter and prove a determining factor, since the larger a plant is, the greater becomes the absolute importance of small economies.

So much for the position of the cooling tower from a commercial viewpoint in its present-day development. Granting the position of the cooling tower and its duty as well defined, many questions arise as to the absolute values of the results attainable and to the reliability or uniformity of the results attained under wide variations of temperature and hygrometric conditions. A consideration of this question brings us into close contact with the ultimate theory of action of the cooling tower itself, and here it can be said that absolute values along these lines have not as yet been obtained. A cooling tower depends for its action upon the cooling effect of the air on the water, and this is accomplished through two entirely different agencies. Water can be cooled by air in direct contact with it, if the air is cooler than the water, by the ordinary processes of conduction and convection. This is the agency that performs the cooling duty in the closed type of cooling tower to a much greater extent than is imagined. The cooler the air is, under these circumstances, the more efficient is the tower in the performance of its duty. A closed type of tower operates under a forced draft, induced by means of fans, and the air almost immediately on entrance to the tower, loses its hygrometric cooling effect, since water is vaporized into dry air with astonishing rapidity.

The hygrometric cooling effect, so called, depends upon the absorptive power of dry air for water vapor when brought in contact with a water surface and the consequent cooling effect upon the water due to the evaporation of a portion of it and the abstraction of the latent heat of vaporization, which is required for this event, from the remaining water. The absorptive power of air for water vapor depends not upon the air itself, but upon the water vapor present in the air at the time and upon its temperature and pressure. The air acts really as an agent for the conveyance of the water vapor present in it, and is important only as its temperature and pressure affect the temperature and pressure of the water vapor conveyed in its midst. This action, however, is quite extensive, since the two materials are quite intimately mixed, and any variation in the temperature and pressure of the one immediately produces a pronounced effect upon the other, due largely to the latent heat capacity of the water vapor when saturated and subjected to small temperature variations.

For a simple understanding of the fundamental problem, the air should be eliminated, or not considered as present during the cooling process as carried on in the tower. The air is actually eliminated in one or two commercial processes involving the same phenomena, with much more pronounced cooling effect. This is actually the case in the so-called vacuum ice-making process, where the vaporization is accomplished by forced variation in the pressure and degree of saturation of the water vapor, with a consequent cooling effect of the water sufficient to transform it into ice. The cooling tower, therefore, is not burdened by the performance of a large duty, from a cooling point of view, at least when the possibilities are considered in other developments utilizing the same agency. However, conditions are essentially different to the extent that the conditions are forced or produced by mechanical means in the one case, whereas natural conditions are used in the other. The utilization of forced draft in the development of the closed type of tower, is a step in the direction of utilizing mechanical agencies in the development of the cooling tower for increased efficiency, and it is an interesting question in regard to the extent to which such developments can be carried with this object in view.

Nothing has been attempted in this line, except the forced-draft development, and this has undoubtedly resulted in a diminished mechanical efficiency, although it has placed the cooling tower of the closed type upon a much higher plane, at least from a reliability and commercial-efficiency standard.

The fact is that ordinary air loses its absorptive power for water vapor within a distance of from 2 to 4 feet after entrance into the cooling tower and from then on it produces its cooling effect solely and simply by conduction, convection and radiation. Saturated water vapor exists in the air in amounts ranging from several grains to the cubic foot at freezing temperatures to an amount as high as 250 grains to the same space at the temperature of boiling water. Its amount is dependent absolutely on the temperature, and its absorptive power at any temperature for water vapor is almost directly proportional to the amount of vapor it possesses at any temperature, to that which it should possess at that temperature if it were saturated. Thus it can be readily seen that ordinary air, under any hygrometric conditions, possesses an enormous capacity for absorbing water vapor, provided only that it is heated sufficiently, and in no case need this exceed that of boiling water. The presence of the air does slightly influence the amount of water vapor present in the space occupied by it, but its effect is comparatively slight in comparison to that of the other factors, and it can be neglected in most calculations in this work.

Thus it can be readily seen that these two cooling agencies act in some cases in direct opposition in their effect in the cooling tower. The cooling effect, due to absorption of water vapor, is proportional more or less directly to the temperature, and the higher the temperature the less is the effect due to the absorption of heat by the processes of conduction, convection and radiation. In some extreme cases, notably of the closed-tower type, where the amount of air furnished can be regulated at will, the cooling effect, due to absorption of water vapor alone, can often be carried so far that the water will leave the tower at a temperature from 25 to 30 degrees below that of the air which produced the cooling effect. Again, it can be said, that the effect due to conduction and convection varies slightly with the temperature, whereas the variation in absorptive power for water varies greatly with the temperature. Hence, the hotter the air is, the greater will be its cooling effect due to absorption, and the less its effect due to conduction, etc. In the example shown the cooling action by conduction, at least during a portion of the process, was undoubtedly negative, and the total cooling effect might possibly have been greatly increased, if the conduction had been almost eliminated or the absorption increased by raising the temperature of the ingoing air.

Both of these modifications are comparatively easy to attain in actual practice. The elimination of the conducting feature can be accomplished by reducing the size of the tower, in the closed type, or cutting down the path of the air in contact, or by speeding up the rate at which the air traverses the tower. In the open type of tower, on the other hand, the same result can be accomplished by installing fans for diminishing the time of air transit, since the length of path in this type is quite small in comparison to the closed tower. On the other hand, conduction plays a most important part in the closed type of tower under adverse hygrometric conditions, such as are often met with under normal atmospheric conditions. Thus, when it is raining, or the atmosphere is saturated with moisture, the cooling effect due to absorption of water vapor is absolutely negligible, and the cooling of the condenser water can be accomplished by conduction alone. Of course, this is carried out in practice by speeding up the fans, and thus increasing the amount of air that passes through the water in a given interval of time. Also, air often exists in a condition approximating this, and combinations, in which the air possesses small absorptive power, but quite low temperatures, are common, and by this means the closed type of cooling tower can be operated under all temperature and hygrometric conditions with a degree of uniformity and reliability in the amount of cooling produced that cannot be met with in any of the open types of towers now on the market.

Thus, the question of the possible heating of the air before advent into the cooling tower is of great importance. It can be accomplished with considerable ease in many installations as they exist in actual operation and with minimum changes in construction and operative details. The majority of cooling towers have been placed on the roofs of buildings, in general to economize space and to allow greater access to the air. This is almost invariably the case in installations of the open type of tower, since they depend for their supply of air on natural agencies. However, they are often placed in this location for an entirely different reason. The hygrometric condition of the atmosphere varies at all seasons of the year very greatly with the hight above the surface of the ground and on this account more satisfactory operation is obtained with the cooling-tower installation on the roof of the building, since the moisture content of the air is much less here. If then the air supply is drawn from the interior of the building for such a tower it will invariably be much warmer and hence possess a much greater absorptive power and water-cooling effect. This is quite possible in most power-plant installations, since the air could be readily drawn from the loft above the boiler or engine rooms, and would be well suited for the purpose.

So much for a possible suggestion in regard to an easy economy in the operation of the closed type of cooling tower. Many other interesting possibilities present themselves in regard to constructive details with the object of further increase in efficiency. Of course, the statement that air loses its absorptive power for water vapor in from 2 to 4 feet in its passage through the tower depends on ideal conditions existing in regard to the distribution of the water in the space traversed by the air. These are not met with in practice and the operative distance can be increased to at least three times this amount with satisfaction on account of the poor distribution of the water, the often high but variable speed of the air, and the interaction of these two effects. A wide and uniform distribution of the water is not so necessary in the closed type of tower as in the open one, since the length of air path in the closed type is much greater. In actual practice, a wide and uniform distribution of the air is not always easily attainable. This is especially the case in the open type of tower, since the air currents cannot be regulated at all, and they tend to redistribute the water and further cause it to be piled up on the leeward side of the distributing devices or pans, so that this feature in turn affects the efficiency of the distributing device, and augments the evil. This shows another reason why the open type of tower may prove unreliable in its cooling action even though hygrometric and temperature conditions are favorable. In some types of open towers, in the case of a high wind, fully 60 per cent. of the surface of the distributing pans may be absolutely dry, showing the inefficiency of this device under some circumstances.

The object of this article was, however, not to consider the relative merits of the two types of towers, but to consider the theory of the cooling tower in the abstract. Both towers possess special advantages, at least from a commercial point of view. The open type can be installed at a much lower first cost than can the closed type, and its cost of maintenance, at least in regard to operative expense, is much less. The closed type, on the other hand, is a more permanent structure, and more reliable in the performance of its duty. Sufficient has been said to show the relative positions of the two types of tower now on the market, and further to show that the theory of these has not been fully worked out and that interesting possibilities present themselves for future work in this field, and let us hope that progress will be more rapid than it has been in the past.

The Experience of Ryan and Casey

By T. T. Roe

Bordering on the open prairie on the northwest side of the city, stood the yards and planing mill of Olson & Co. The street in front of the plant was a dirt road, impassable when wet, save by using the flat rails of the horse-car line, over which occasionally passed an ancient car drawn by a team of fat steeds. A few unfenced cottages were scattered about and from these rambled together children, chickens and goats.

The mill was a small affair, employing but a dozen men. The power department was in one narrow room and consisted of a 60-inch by 16-foot boiler, having one wall as part of the building, the opposite one being run up to support the floor above, which was used as a drying room. In front of the boiler and not more than 6 feet from it was a center-crank engine of the single-valve type, exhausting into a small closed heater. A small steam pump completed the outfit. On the right

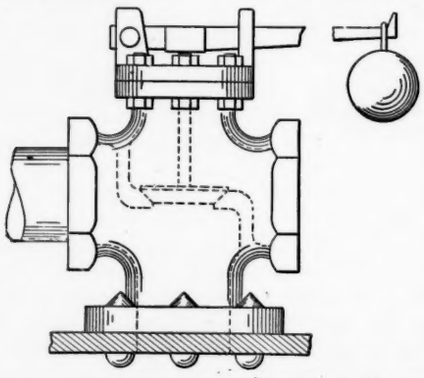


FIG. 1. ORIGINAL BALL AND LEVER VALVE

of the boiler was a drop door opening into a little coal shed. On the left was a glass door that led into a small narrow room provided with a bench and vise and a door opening outside. To reach the top of the boiler one went outside and climbing a rickety ladder opened a 24x24-inch door and crawled over the rear arch. Few had the curiosity to crawl through this opening and encounter the dust of ages. That the plant had run in this manner a number of years without being destroyed by fire was more astonishing than the failure of a butt-strap triple-riveted joint and could be attributed only to an all-powerful streak of luck.

The engineer in charge was a husky old man with a warlike mustache directly in the rear of a black dudeen that defied continually the smoke ordinance. That he was a fine mechanic no one ever denied and, indeed, Ryan himself admitted the fact frequently and casually as one would comment on the weather. He was engineer, superintendent, repair man and the boss of the entire outfit.

The fuel consisted of shavings and the supply was inexhaustible. The operation of firing was very simple and consisted of

opening the large firebox door, taking a big scoop full of shavings and packing the furnace full. If Ryan now and then used a few shovels of stock coal it followed that the shaving pile increased proportionally and with an eye for saving fuel he troubled the coal pile but little.

To this peaceful scene drove Casey, a riveter, and by grace of membership to a political club, a city boiler inspector. The chief had complained about inspections being overdue and Casey, scouting in this section, saw the stack afar off.

At the office of the mill his rude voice scattered the peaceful owners who promptly referred the matter to Ryan. The certificate was found and proved to be three years old. At a glance Casey knew that Ryan would not assist in the "shaking-down" process and loudly demanded that they pull the fire at once or stand a fine, whereupon Ryan proceeded to enter the telephone booth and call up his friend, the mayor, and report this red-haired intruder. This action, as intended, caused Casey to cool down and it was agreed the boiler would be ready for inspection two days later.

On the day set, Casey arrived with a helper as husky as himself. The boiler was cold and filled with water, the pressure used being 70 pounds. The water test was 105 pounds, Ryan attending to the safety valve.

Now, Casey was out for trouble and bound to find something on which to base a kick. But the tubes were tight, the seams and rivets showed no leaks and the fire sheets and heads were good. The brickwork was all right, and the water column, gage cocks, etc., were beyond criticism. The grin of superiority on Ryan's face and the air of tolerance with which he glanced at Casey's endeavors to get back at him caused Casey to determine that a kick would be forthcoming. Failing to find cause, he went to the outer door and climbed the ladder leading to the top. Through clouds of dust, he proceeded to the dome. Here he found a tee ball-and-lever safety valve as shown in Fig. 1. Now, in this city, the law called for spring-loaded safety valves and did not allow ball-and-lever valves to be used. Here was his chance to kick and he loudly proclaimed that if within three days the boiler was not provided with a pop safety valve, Mr. Ryan would have to give up his license. He tore out a slip of paper and wrote out a formal notice, serving it on Ryan in the presence of the owners. Ryan handed the paper to the owner and remarked that he had better throw it away and get the mill going, as they were behind on orders. The blood of Casey boiled and be retorted fiercely, whereupon Ryan told him to remove his shield and he would toss him over the roof. Casey left, vowing he would personally escort Ryan to the workhouse upon his return.

Fig. 2 shows how Ryan converted the 4-inch lever valve into a 4-inch pop valve. The spring was an ordinary draw-bar spring picked up on the railroad. Tension was produced by drawing down on the upper nuts and the valve relieved at

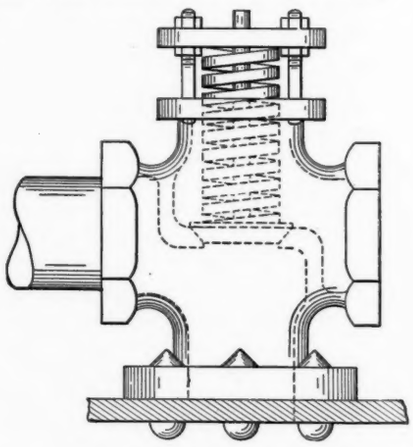


FIG. 2. VALVE AS ALTERED

70 pounds, closing at 66 pounds. The valve was reduced to 3/16 inch where it touched the seat, thus affording prompt closure, which would not have been the case if the lip had been wide.

Upon Casey's return three days later, armed with a warrant to arrest everybody in sight, he was told to see the new pop valve and as he crawled on top, the valve happened to blow off. Finally he viewed the valve and with curses descended. Tired of the man, Ryan threw him through the window into the slack coal pile, relighted his pipe and resumed the work of passing the shavings into the furnace. Casey arose and to his helper's inquiry if he would run Ryan in, responded: "Lave him alone, Dan, he's a bull of a man."

Coal and Ash Handling at Large Plant

By Warren O. Rogers

The Birmingham Railroad, Light and Power Company has equipped its power plant with an unusually complete system for handling the coal and ash for the 17,000 horsepower of boilers installed. The plant is one of the largest in the South.

The method of handling coal and ash at the power plant of the Birmingham Railroad, Light and Power Company is somewhat different from that used in any other large station, as far as the writer is aware. The thirty-one boilers, aggregating over 17,000 boiler-horsepower, naturally consume considerable coal and consequently produce a large amount of ash.

Fig. 1 shows the layout of the plant in cross-sectional view, also the ash tunnel between the two rows of boilers and the coal pockets on the outside of the boiler house. Fig. 2 shows a view of the coal-handling apparatus. The coal is delivered in the regulation-size coal-carrying car and emptied through the bottom hoppers into the coal pockets, which are arranged as shown at *A*, Fig. 1. These pockets extend the entire length of the boiler room and each pocket is provided with a gate operated by a handle from a walk-way alongside the belt conveyer.

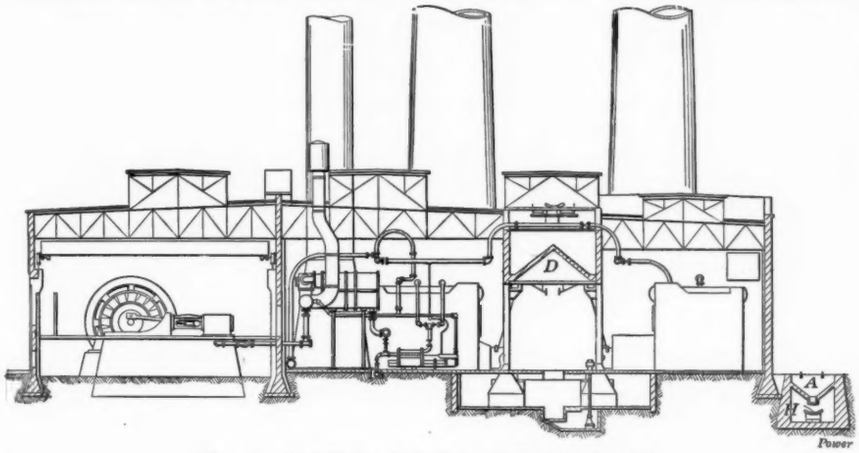


FIG. 1. VERTICAL SECTION THROUGH PLANT

Under these pockets is a belt coal conveyer, which takes the coal away from the pockets as the attendant allows it to feed through the bottom. The coal is delivered to a set of crushing rolls and passes through them into a small concrete storage bin. The outlet of this bin is fitted with a swing gate, which fits in the trough *B*, Fig. 2, and is operated by the hoisting engineer who is stationed at *C*.

Over the boilers is located the main coal bin, the construction being of concrete, with the bottom shaped as shown at *D*, Fig. 1, so that the coal supply will freely pass to the coal chute running to each stoker. Over the bin is placed a belt conveyer, operated by a motor; above the conveyer at the receiving end is a second set of crushing rolls.

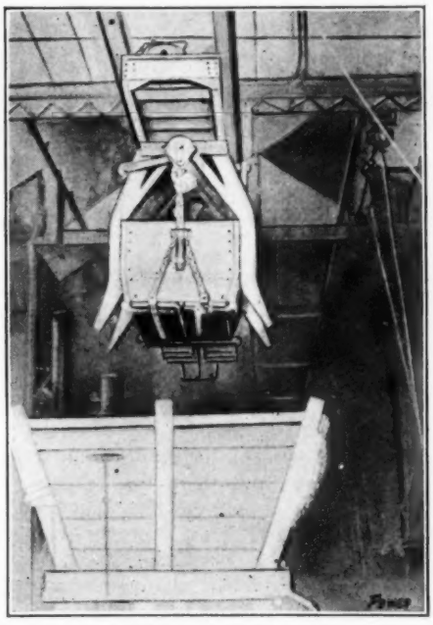


FIG. 3. ASH SKIP

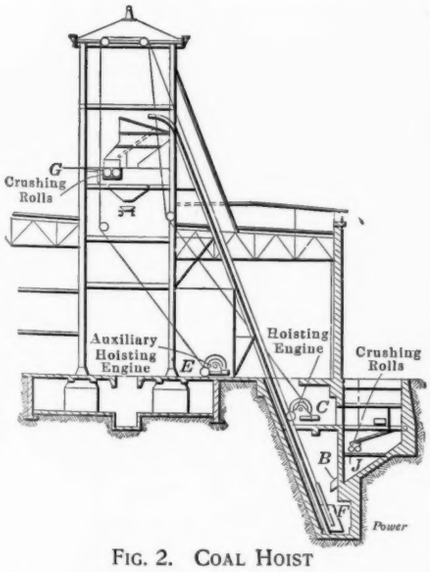


FIG. 2. COAL HOIST

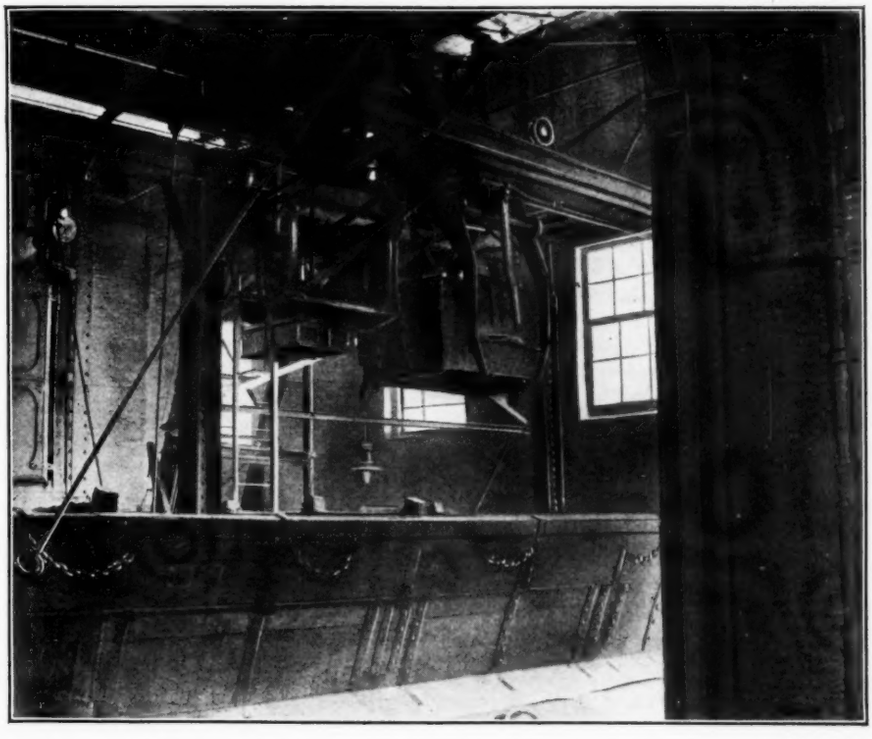


FIG. 4. ASH CAR, SKIP AND TRAVELING ASH HOIST

From a point above the crushing rolls, *G*, Fig. 2, to a point in the pit, is an incline hoist on which a coal skip is operated by means of a steam hoisting engine, shown at *C*. An auxiliary hoisting engine is placed at *E*.

When coal is required in the upper bin the operator starts the underground conveyer *H*, also the lower crushing rolls *J*, with the skip *F* in its lowest position, as shown in Fig. 2. The gate of the chute *B* is then opened, which allows the coal to fill the skip. The gate is then closed and the skip of coal hoisted to the top crusher rolls *G*, which can be used or not, depending upon whether the lower rolls *J* are in working order or not. The coal, after being delivered to the top conveying belt, is carried to the proper point over the coal bin and discharged where desired. The upper conveyer is housed on the roof of the boiler house, as shown in Fig. 2.

HANDLING THE ASH

The method of handling the ash is also out of the ordinary. In the center of the boiler room is placed a standard-gage track, which serves a double purpose, that of allowing loaded freight cars and empty ash cars to run into the boiler room for the removal of ashes.

Between the rails of this track are open spaces which permit hoisting the ash skip from the basement by means of an overhead traveling hoist. Beneath the boiler-room flooring is a sub-basement. Two rows of posts support the track and boiler-room floor. Between these posts is a trench in which the empty skips are lowered ready for filling.

The ashes from the boiler furnaces fall into an ashpit, the bottom of which is at such a hight that a common ash barrel can be wheeled alongside and loaded with ashes, which are hoed out by hand. The ashes are then dumped into the nearest skip, of which there are a number, and when full the operating boy with his traveling hoist lifts the skip by means of hooks which engage in the chains shown at each end of the skip, Fig. 4. The skip is held in place by arms, which engage with lugs, one arm being placed on each side of the skip. This prevents accident to the workmen as the loaded skip is hoisted and conveyed from one end of the boiler room to the dumping point. The four legs of the ash conveyer, shown in Fig. 3, are for the purpose of steadying the loaded ash skip and also to cause it to always hoist into a certain position so that the catch arms and lugs will always engage with each other. When the load of ash is hoisted and secured in position, it is run to the far end of the boiler room, where ash cars are waiting, Fig. 3. Fig. 4 shows a side view of the hoist, skip and ash car.

Natural Draft and Chimney Design*

By H. G. Brinkerhoff

A consideration of the factors affecting the principal dimensions of chimneys, in which a few prevailing erroneous impressions are corrected. A number of constants used in practical chimney design are also given.

Nearly every treatise on chimneys and natural draft contains the following statement: "The natural draft obtainable from a chimney depends upon the column of gas inside the chimney being lighter than the air outside." This, however, is not the only cause in producing draft, as a wind blowing over the chimney produces a considerable suction without the inside air or gas being heated at all. This fact is important, but as the wind is such an uncertain element, it cannot be included as a factor when calculating the dimensions of a chimney.

There are two principal dimensions to be determined, the hight and the cross-sectional area of the chimney. A proper hight is of the greatest importance as this produces the pulling force, which must be sufficient to overcome the friction through the grates, flues and chimney.

A hight sufficient to produce a strong, vigorous draft involves considerable expense, but this can be offset by the use of fewer boilers. It is becoming to be appreciated generally that there is economy in running with fewer boilers due to the following causes: There is less expense involved in banking the fires at night; with an intense fire the gases are more completely burned to CO_2; a better circulation is set up; and less smoke is produced.

It has come about in practice that the purchaser wishing to secure 1000 boiler horsepower is furnished with 10,000 square feet of heating surface or ten square feet per horsepower, which is a safe factor to use with ordinary draft. But whether the purchaser obtains 1000 horsepower or twice that amount is purely a matter of supplying heat enough which, in turn, is practically limited only by the draft necessary to burn the requisite amount of coal.

The boiler horsepower does not need to be considered when determining the hight of a chimney, although with rare exception this is now erroneously being done. It is universally taught that draft varies with the square root of the hight, but this use of the word "draft" applies only to the capacity, which is largely a matter of area. Chimneys produce a draft which is directly proportional to their hight, that is, a 200-foot chimney will produce a draft pressure which is twice that of one 100 feet high.

Many papers have been written on the subjects of chimneys and drafts, generally developing abstruse or speculative theories. The most helpful and practical one that I have come across is an article by T. F. J. Maguire, a summary of which is as follows:

The loss of draft in boiler settings is placed at 0.3 inch for water-tube boilers, increasing to 0.4 inch with 50 per cent. overload. Of course, this is intended to be used with judgment to suit the construction and baffling of the particular boiler under consideration. The writer believes that horizontal return-tubular boilers would require the same allowances, and the vertical types about 0.15 inch.

Maguire allows 0.1 inch per 100 feet of length for circular iron flues and 0.05 inch for each right-angled turn. If square or rectangular in shape, the foregoing should be increased 25 per cent.; for brick flues it should be increased 30 per cent.

Let these factors be applied to a concrete case where 25 pounds of soft coal

DRAFT PRESSURES REQUIRED IN FURNACE.

Kind of Coal.	Pounds of Dry Coal Burned per Square Foot of Grate per Hour.						
	15	20	25	30	35	40	45
	Draft in Inches of Water.						
Eastern bituminous coals	0.12	0.16	0.20	0.27	0.34	0.42	0.52
Western bituminous coals	0.15	0.20	0.25	0.33	0.42	0.52	0.65
Semi-bituminous coals	0.15	0.20	0.28	0.37	0.48	0.60	0.80
Anthracite buckwheat, No. 1 and larger	0.45	0.70	1.00				
Anthracite buckwheat, No. 2 and No. 3	0.75	1.30					

*Abstract from paper delivered before the National Association of Cotton Manufacturers, at Portsmouth, N. H., September 17, 1910.

are burned under horizontal return-tubular boilers with two right-angled turns in the breeching.

Furnace draft for 25 pounds soft coal	0.2
Horizontal return-tubular boilers*	0.3
Fifty feet of steel breeching	0.05
Rectangular shape requiring 25 per cent. additional	0.0125
Two right-angled turns	0.1
Round-brick chimney 150 feet	0.15
Brick surface 30 per cent. additional	0.045
Total draft required	0.8575

*The boiler resistance increases with the velocity. Therefore, if 25 pounds per square foot of grate area develops a considerable amount above rating, add a third to the boiler resistance. In this case it would be 0.4 for the horizontal return-tubular boilers instead of 0.3.

This is approximately 0.9 inch, and as 0.6 inch can be obtained with 100 feet of stack, then a 150-foot stack would provide the required pressure.

It may be pointed out that no allowance has been made for the effect which an economizer would have on the draft. But as the economizers with straight open passages usually have a greater free area than that of the flue, it can be counted as flue length.

The selection of a suitable area for a chimney is as frequently a matter of guesswork as the determination of the hight. Some have a pet rule of making the cross-sectional area proportional to the grate area regardless of the fact that if twice as much coal were burned in one case as in the other, it would call for twice the flue area, because of one having double the volume of gas. As a high chimney costs so much more in proportion than one of large area it is common fallacy to curtail on the former and increase the latter. Thus, for 1500 horsepower the owner provides himself with a stack of a hight suitable for only 1000 horsepower and uses an area good for two or three times this amount. Then he wonders why he has difficulty in getting 1500 horsepower and has to have recourse to artificial draft. The theoretical area of chimney required for 1000 horsepower with a stack only 100 feet high is 7 square feet under ordinary conditions. In my own practice, I usually allow 20 square feet per 1000 horsepower of boiler installation. For 2000 or more horsepower I reduce this allowance about 10 per cent., increasing the reduction on very large plants to perhaps 20 per cent., and keep 16 square feet as a safe limit. For smaller plants, say, 500 horsepower, I would allow 12 square feet, which is then more than three times the theoretical area on the moderate basis of a hight of 100 feet.

If the chimney is over 100 feet in hight, the theoretical requirement of 7 square feet is reduced, but I never use less than 16 square feet of area.

The least allowable area should give the best draft. Were the 20 square feet per 1000 horsepower to be increased to 30 square feet it would mean, in a travel of 150 feet up the stack, together with a flue length of 30 feet to reach its port, that there is 645 square feet of additional surface to be heated to the same temperature as the gases with exposure to radiation loss, as well as increasing the air leaks.

An unutilized excess of draft pressure is bad and the boilers should be fired to as near the limit as possible, but at times there will be necessarily a surplus suction produced by the wind. That this excess is common is shown by the operation of the automatic regulation of dampers which continually cuts off 25 to 50 per cent. of the available draft throughout the entire day. Of course, the damper regulation effects a saving in preventing an undue excess of air through the grates, but it should, as far as possible, produce a more intense fire and by heavier coaling enable the plant to run with fewer boilers.

In the matter of draft regulation, it is much better to have a regulator that operates slowly than to have one very sensitive which will open and close on a slight variation of pressure, thereby letting in an excess of air one minute and cutting off the air supply the next minute. With such a sensitive damper, it is only necessary to limit the range of damper opening, so that it does not shut tightly nor open to its fullest extent. The regulating damper should be as near the chimney as possible, and if it were placed at the top of the chimney, it would be even better, as it would then hold back that much more heat. As operated in many plants, when the damper is closed at the port, the brick lining or core loses much of its stored-up heat and the chimney cools off; then when the demand comes for more steam, it is some time before the chimney recovers enough lost heat to produce a vigorous draft. The condition becomes worse where the regulation is applied to dampers in the boiler uptakes, for during the time the damper is closed the flues have cooled as well as the chimney.

The Economical Use of Lubricants

By William M. Davis

The efficient and economical use of lubricants depends principally upon the selection of the proper lubricant for the particular service in which it is to be used, obtaining the lubricants at the lowest market price and a well ordered system of handling, distributing and applying the lubricant.

Lubricants may be divided into three general classes according to their forms: fluid, plastic and solid. To the first class belong the various oils, to the second class the greases, and to the third such substances as graphite, soapstone, mica, etc.

Where the speed is high and the pressure great, oils are the most satisfactory. They form an elastic coating to the metal which keeps the contact surfaces apart and carries away the frictional heat. Another advantage is that they can be had in almost any desired grade or density, and do not become rancid nor gummy. In the early days of engineering, the oils were all of animal or vegetable origin. These, although good lubricants, had their disadvantages as they were expensive and had a tendency to become rancid. In recent years these have been almost entirely displaced by mineral oils which are, in many respects, better lubricants and are not subject to the disadvantages of the animal or vegetable oils.

The greases are more suitable on slow-moving machinery where the pressure is not great, but if used indiscriminately they will produce a noticeable increase in the friction load.

Greases may be divided into two classes: the lime and potash soaps which have a high melting point and those of a tallow base which have a low melting point. The first are made by forming a small amount of fatty oil into a soap by means of lime water, caustic potash or other alkali, and mixing it with a large amount of petroleum oil. Such greases may be made in any degree of density and have a melting point of from 140 to 180 degrees Fahrenheit. The greases having a tallow base are composed of a large percentage of tallow combined with an alkali and are brought to the desired density by means of vaseline-petroleum oils. These greases have a low melting point, usually from 116 to 120 degrees Fahrenheit.

The solid lubricants, such as graphite, soapstone, etc., have a limited field of use, being used principally as a filler for fibrous piston-rod packings, etc.; although a certain form of graphite has shown in laboratory tests to have a high lubricat-

*Abstract from paper delivered before the National Association of Cotton Manufacturers, September 15, 1910.

ing value with a low coefficient of friction.

There are three essentials to be looked after in obtaining economical lubrication; first, the selection of such lubricants as will maintain cool bearings, have a low friction loss, etc.; second, obtaining these lubricants at the lowest market price; third, their economical use.

The selection of proper lubricants depends upon the class of machinery on which they are to be used. On light-running and high-speed machinery the light-bodied oils will give the best results; for slow-speed machinery, the heavier oils will be better. On slow-speed engines where the oil is fed from cups, a heavy oil should be used; and for high-speed engines, where continuous oiling systems are employed, a light-bodied oil should be used.

Where the plant is small, the most satisfactory method of purchasing lubricants is to buy from some reliable oil firm, but where the cost of lubrication runs into thousands of dollars, as is the case with many large companies, the best plan is to purchase on specifications, stating clearly just what is needed and awarding a contract for a year's supply to the lowest responsible bidder. Specification buying is fair both to the consumer and to the dealer.

The usual physical tests for gravity, flash point and viscosity, and in the case of cylinder oils, a chemical analysis for the percentage of fatty oil will give anyone who is familiar with lubricants a very close indication of the relative values of different oils. If one wishes to go further it is often possible to make actual service tests which will show any difference in the lubricating property of two or more oils. If it is desired to make a service test, select a bearing which is running under constant load and speed. Place a thermometer in the bearing so that the bulb rests on the shaft, and maintain a constant feed of oil. Have another thermometer placed in the room near the bearing, but out of any drafts, and note the room temperature.

Commence the test when the engine is started, note the rise of temperature at frequent intervals, and continue until the temperature of the bearing ceases to rise. Every bearing will in the course of a few hours reach a point where the heat is radiated as fast as it is generated. Deducting the temperature of the room from that of the bearing will give the temperature due to friction. If the engine runs in the daytime only, the bearing will cool off during the night; then repeat the test the next day with another oil. When making such tests it is well, after finishing one test and before commencing another, to wash out the bearing with gasolene.

Although it has been found in experimental work that the coefficient of friction often decreases with a rise in temperature, yet in every-day practice it is safe to assume that the oil which will keep the bearings the coolest is the best lubricant.

If it is desired to ascertain the lubricating value of two or more cylinder oils, take one oil and feed it at a given rate for a few days, then remove the cylinder head and wipe the surface with a piece of soft white paper. If a stain of oil is found it is evidence of good lubrication; if there is no stain and a liberal amount of oil has been used, it indicates either that the steam is very wet or that not enough fatty oil has been used in compounding the lubricant.

Next to the problem of buying, efficient and economical lubrication depends upon the methods of handling and applying the lubricant. It is a waste of money to fit engines with a continuous oiling system unless necessary precautions are taken to recover the oil used.

Another cause for loss often occurs in wiping up with waste. In most large plants it will pay to install some make of oil- and waste-saving machine, by means of which the oil is extracted and filtered and the waste washed, dried and used over again.

Receiving, storing and distributing lubricants are other important factors. If the plant is a large one, it will pay to provide storage capacity so that the oils can be bought in tank-car lots. But if it be a small plant where only a few barrels of each kind of oil are used per month, the oil should be kept in tanks so arranged that the barrels can be emptied into them by gravity. Care should be taken to see that the barrels drain out thoroughly.

It is customary in a large plant to have someone in charge of the oil house, to receive and store the lubricants, and to deliver them to the various engine rooms and departments, keeping a record of the amounts distributed.

No oil should be issued except on a requisition signed by the chief engineer, master mechanic or department foreman. In a small plant where the amount used does not warrant keeping a man especially to look after the lubricants, the oil house may be placed in charge of the general storekeeper.

At the end of each month the quantities of lubricants issued should be totaled, and their cost figured and charged to the individual departments using the oil. By comparing monthly reports, it can be seen at a glance whether the oil bill is increasing or decreasing and in which departments the differences have taken place.

New Power Plants for San Diego, Cal.

The San Diego Consolidated Gas and Electric Company, of San Diego, Cal., is adding to its plant four batteries of 500-horsepower Babcock & Wilcox boilers, which are equipped with the latest improvements and will furnish steam for a 2000-kilowatt Curtis turbine of the horizontal type driving a General Electric alternating-current generator. A portion of the old engine house is now being taken down and a new one erected, which will be equipped with an electrically operated traveling crane. Provision is being made for the extension of the power plant in the near future. It is intended to put in a double tunnel for the circulating-water intake and discharge. The tunnels are to be 5 feet in diameter and built of reinforced concrete. The discharge tunnel will be on top of the intake tunnel and both will be under the low-tide mark and about 1500 feet in length. The foundations for the turbine and auxiliaries and the engine house are now being put in, and the work is being done under the personal superintendence of H. M. Reilley, construction engineer, of Billesby & Co., of Chicago.

The San Diego Electric Company are also having plans drawn for the construction of a new power station to cost ultimately about $225,000. The structure will be one of the most complete and uptodate of the kind on the Pacific coast.

When completed, the new power plant will be about 200 feet in width by 300 feet in length. It will cover an area approximating 60,000 square feet, and will generate about 35,000 kilowatts.

The boiler room will have eight batteries of six oil-burning boilers each, which will require three large smokestacks. Glass and louvered skylights will furnish light and air to the workmen below. The engine room will be 75 feet in width by 300 feet in length, and from floor to floor will measure more than 60 feet. It will be amply lighted by six windows, each 50 feet wide and 50 feet high. An electrically controlled traveling crane, having a lifting capacity of 65 tons will run the full length of the room. A 10-ton crane located under the switchboard gallery will care for all motor-generator machinery. The switchboard gallery will be above the entire floor and will afford a clear view of the room at all times. As soon as the working plans have been completed the actual work on the building will be begun.

High Speed Compound Engine Design

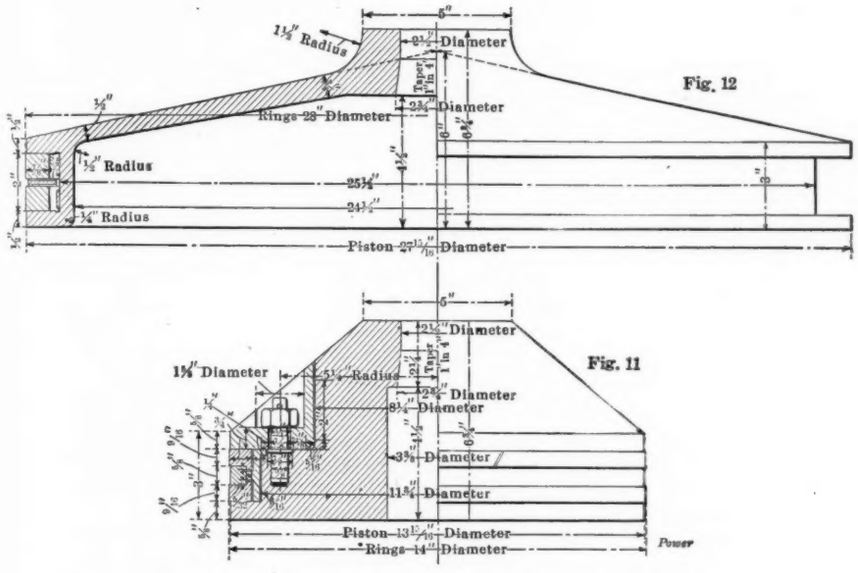
By J. D. Speedium

The second instalment of a series of articles on High-Speed Engine Design in which the pistons, valves, guides, piston rods, cross-heads, connecting rods and valve gears are considered.

Pistons

The details of the high- and low-pressure pistons are shown in Figs. 11 and 12, respectively. As is the usual custom the high-pressure piston is fitted with junk rings, and is made of cast iron. The low-pressure piston is of wrought steel having a tensile strength of not less than from 76,000 to 84,000 pounds per square inch. For the latter piston, one type of ring is universally used, and is known as the "Rowan" ring; this is shown in Fig. 12, and consists of two ordinary ramsbottom-type rings of cast iron, turned to the exact size of the cylinder. These two rings are fitted into one wide groove, and between them is a steel wave spring which keeps them against the piston flanges. This prevents rattling and rapid side wear so often noticeable in high-speed engines when fitted with ordinary ramsbottom rings, unless made very deep in section. Behind each ring are placed other wave springs, which press the rings against the cylinder walls. As the pressures outward and sideways are adjustable independently of each other, the arrangement makes a very efficient piston packing for all low-pressure pistons; but for very high pressures and particularly where superheated steam is used, a restrained type of ring is absolutely necessary. The "Rowan" type of packing ring does not require a junk ring on the piston. This tends toward simplicity, and makes a light piston, but as it is necessary to make the weight of the high-pressure piston equal to that of the low pressure, for balancing purposes, there is no difficulty in arranging for a junk ring on the former. In the case of triple-expansion engines the intermediate pistons are usually fitted with rings of the "Rowan" type and no junk plates are fitted.

The designs of pistons here shown represent general practice. The low-pressure piston being of steel can with safety be made very thin, particularly as it is conical in shape. This makes a very strong form, and at the same time assists drainage. Wrought steel is the material generally used for these pistons, but cast steel is coming into common use. The following table gives the usual thickness of wrought-steel pistons:



FIGS. 11 AND 12. HIGH- AND LOW-PRESSURE PISTONS

TABLE 4. THICKNESS OF WROUGHT STEEL PISTONS.

Diameter of Piston.	Thickness of Plate near Boss.	Thickness of Plate near Outer Edge.
16 inches	$\frac{9}{16}$ inch	$\frac{5}{16}$ inch
20 inches	$\frac{5}{8}$ inch	$\frac{3}{8}$ inch
24 inches	$\frac{11}{16}$ inch	$\frac{7}{16}$ inch
30 inches	$\frac{3}{4}$ inch	$\frac{1}{2}$ inch
36 inches	$\frac{13}{16}$ inch	$\frac{9}{16}$ inch
48 inches	$1\frac{1}{16}$ inch	$\frac{11}{16}$ inch
60 inches	$1\frac{1}{8}$ inch	$\frac{3}{4}$ inch

High-pressure pistons are usually of a conical form, the depth overall being made the same as that of the low-pressure piston to enable piston rods of the same length to be used for both high- and low-pressure cylinders. To prevent the possibility of studs working loose, they are fitted with collars, or carried through the piston and riveted over on the under side, the nuts being secured by means of split pins. Packing rings of many types are in use, but the only successful ones are those of the restrained type. The design shown in Fig. 11, and known as the "Admiralty" ring, is extensively used and is simple. In this design the spring rings are prevented from being steam expanded, which causes excessive friction and wear, by lips which engage with a solid bull ring. As a junk ring is fitted, the spring rings can be made of deep

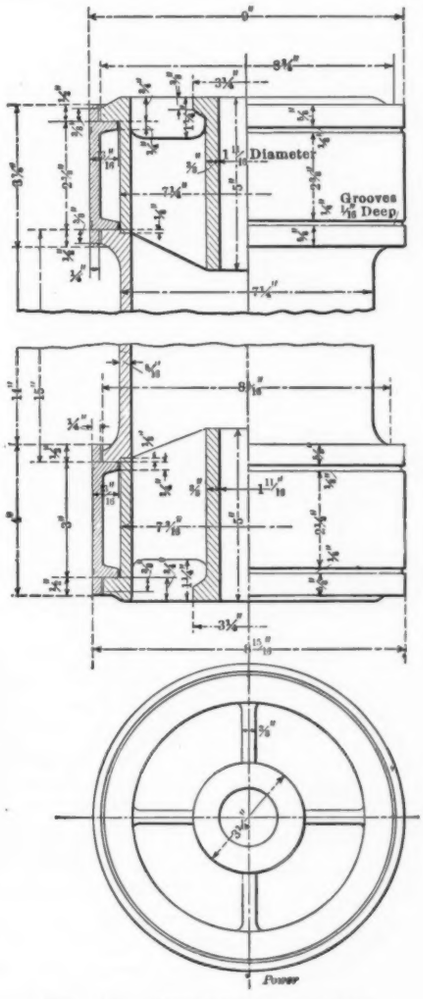


FIG. 13. HIGH-PRESSURE VALVE

section; consequently they will wear for some time before adjustment is necessary, and even then the junk plate can be closed a little to take up the play.

Piston Valves

The type of piston valve most commonly used is the one fitted with solid rings as shown in the detail drawings, Figs. 13 and 14. This design makes a very simple, light and frictionless valve, which is

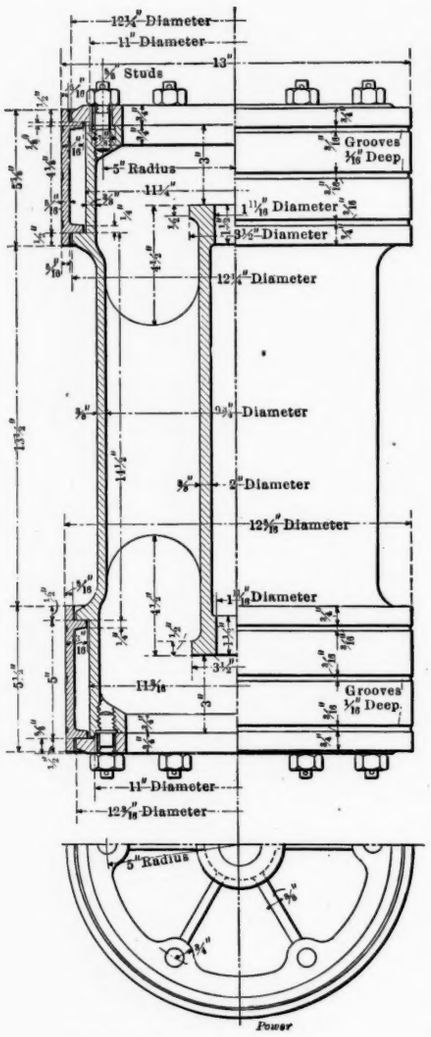


FIG. 14. LOW-PRESSURE VALVE

suitable for any condition as regards pressure or temperature of the steam. It may appear at first sight to be a very uneconomical design of valve, and also one which would readily wear and allow excessive leakage to take place. That might be the case if the engines were of the horizontal type, and the valves worked in a horizontal position, but as high-speed engines are all of the vertical type, and must of necessity be accurately built, the solid rings prove very efficient. It is, of course, essential to have the valve gear truly alined, as side pressure on the valves would cause wear. Solid valve rings will work for several years without appreciable wear, often not exceeding 0.003 inch after being in service for four or five years.

Fig. 13 represents the high-pressure valve. The body is built up in three pieces, held together by the valve spindle, sufficient clearance being allowed in the grooves for the rings to float without being slack. Both rings and valve are made as light as possible, which reduces the effect of inertia and relieves the valve motion. It will be noticed that lips are shown at both ends of the valve rings. These provide a ready means for altering the steam or exhaust lap without making any structural changes in the valve body. Valves are made of similar design to that shown in Fig. 13 in sizes up to about 12 inches diameter.

In Fig. 14 is shown the low-pressure valve. In this case the valve body is made in one casting, loose junk plates being fitted at each end to secure the valve rings. Studs are used for securing the junk plates, and they are locked in a similar manner to those on the high-pressure piston.

As the exhaust steam has to pass through the center of the valves, the diameter is usually determined by the area required for the steam. This should always be at least 10 per cent. in excess of the port area.

Grooves are turned in the rings as shown, and are necessary to retain the lubricant.

VALVE DIAGRAMS

In Fig. 15 are shown the valve diagrams for determining the necessary laps and angle of the eccentrics to give the required cutoff and compression, etc. Referring to the high-pressure diagram, it has already been shown that a mean cutoff of 61.5 per cent. is required to develop the maximum load; therefore the valve must be set to suit. As previously explained, it has been found advantageous for economy, to set the valve with negative lead, so as to obtain a late release with a reasonable compression. In this case a negative lead of ¼ inch top and ⅛ inch bottom will be found satisfactory, and a travel of 5 inches will give sufficient port opening. Roughly speaking, a travel equal to twice the width of the steam port will be found sufficient. The speed of steam through the slots in the liner at minimum opening to steam should not exceed:

High-pressure cylinder . . .	8,000 feet per minute
Intermediate-pressure cylinder	9,000 feet per minute
Low-pressure cylinder	10,000 feet per minute

In the case of compound engines the speed for low-pressure cylinders should be taken as above for the intermediate cylinders.

For exhaust, the valve usually fully opens the port; consequently the speed

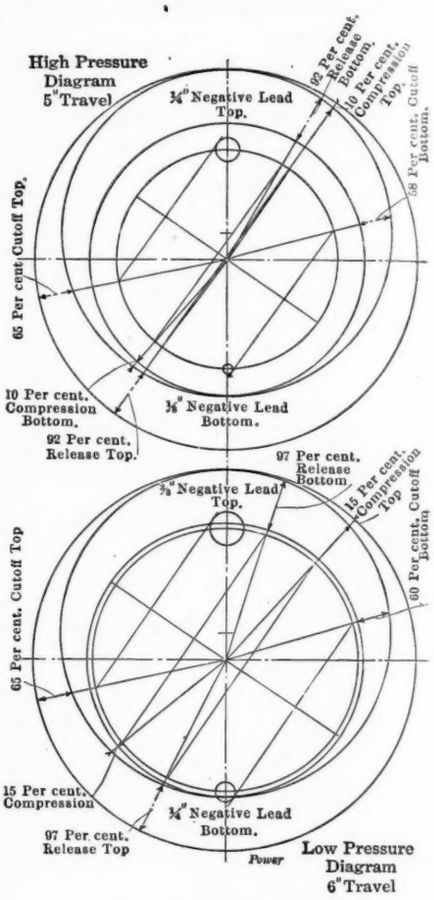
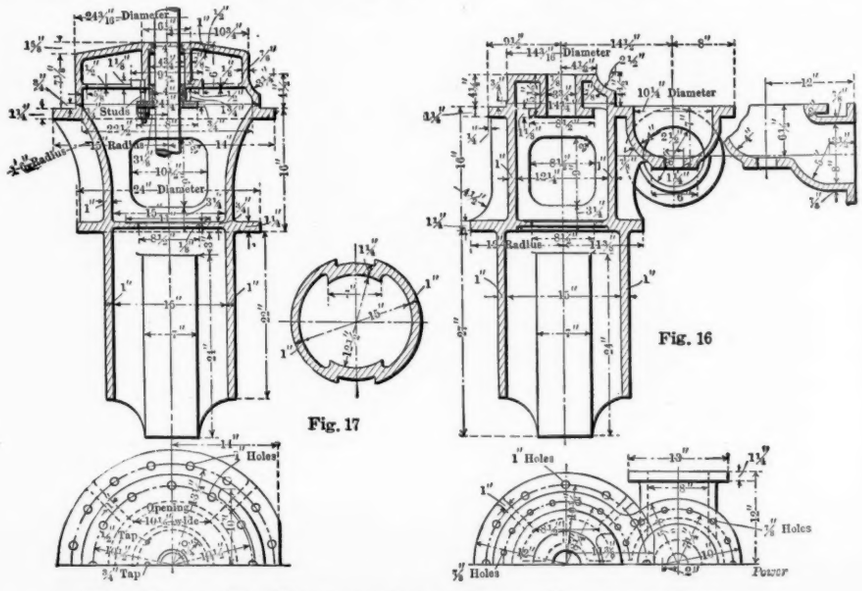

High Pressure Diagram 5" Travel
¼" Negative Lead Top.
92 Per cent. Release Bottom.
10 Per cent. Compression Top.
68 Per cent. Cutoff Bottom.
65 Per cent. Cutoff Top.
10 Per cent. Compression Bottom.
⅛" Negative Lead Bottom.
92 Per cent. Release Top.
97 Per cent. Release Bottom
⅛" Negative Lead Top.
15 Per cent. Compression Top
60 Per cent. Cutoff Bottom
65 Per cent. Cutoff Top
15 Per cent. Compression
¼" Negative Lead Bottom.
97 Per cent. Release Top
Power
Low Pressure Diagram 6" Travel

FIG. 15. VALVE DIAGRAMS

through the slots in the liner is less than through the cylinder ports.

A compression of about 10 per cent. is required in the high-pressure cylinder, and as will be seen from the dia-



FIGS. 16 AND 17. HIGH- AND LOW-PRESSURE CROSSHEAD GUIDES

gram, this gives a release at 92 per cent., which is very satisfactory. The steam and exhaust laps can now be measured and the valve design completed.

In the case of the low-pressure diagram a travel of 6 inches will be found necessary, and to give approximately an equal distribution of power between the two cylinders a cutoff of about 62.5 per cent. will be required. Negative leads of 3/8 inch at top and 1/4 inch at bottom will be suitable, and as the engine is condensing, a mean compression of 15 per cent. may be allowed, which gives release at 97 per cent. of the stroke. It is always a convenience to draw the valve and piston diagrams in conjunction with one another as shown in Fig. 15, as the setting can be read off at a glance.

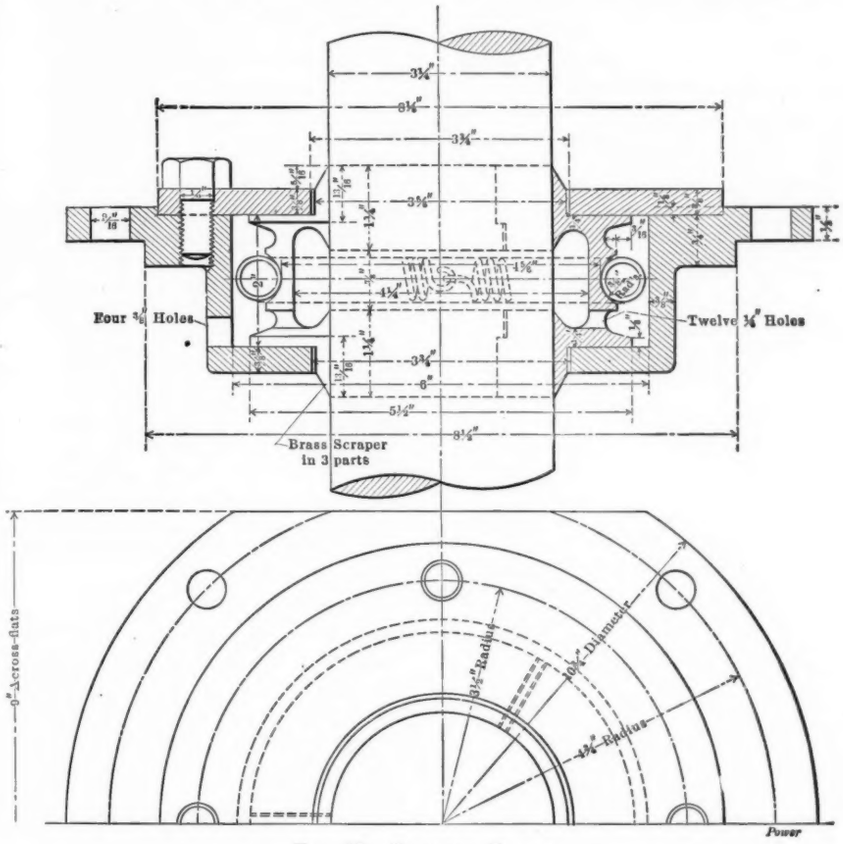


FIG. 18. SCRAPER GLAND

CROSSHEAD GUIDES AND DISTANCE-PIECE CASTINGS

In Fig. 16 is shown the distance-piece casting which supports the high-pressure cylinder, and also forms the bottom cylinder cover. It will be seen that the high-pressure exhaust branch is made in one with this casting, making a very compact and neat arrangement. This branch has to assume a rather peculiar shape in order to make room for the valve-spindle stuffing box above the top of the frame. With the bottom cover is formed a flange to which is attached the metallic packing.

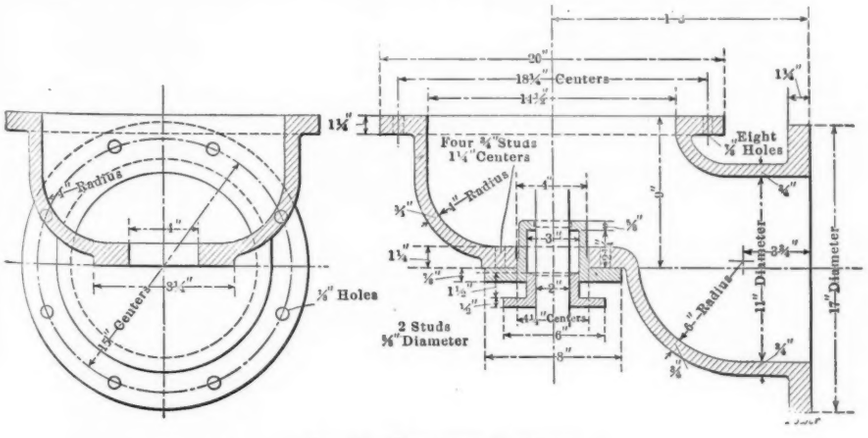


FIG. 19. EXHAUST BRANCH

The crosshead guides are of the bored type and of such a length as to allow the slippers to overrun 2 inches at each end of the stroke. The tubular section makes a very stiff construction, and as the cylinder-cover portion, which forms a spigot fitting the cylinder barrel, is machined from the bored guide, the guide is self-centering with the cylinder. On a level with the top of the frame is fitted another stuffing box or scraper gland, for the purpose of preventing water from the glands getting down into the crank chamber, and also to prevent oil from the crank chamber passing up the piston

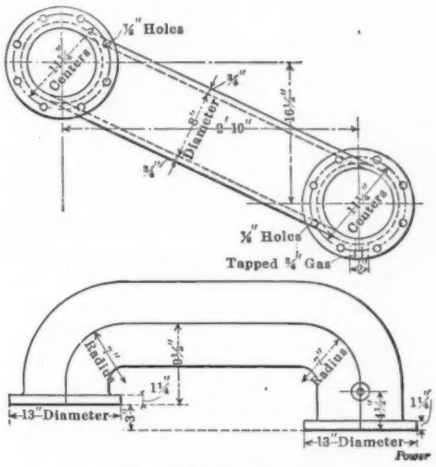


FIG. 20. RECEIVER

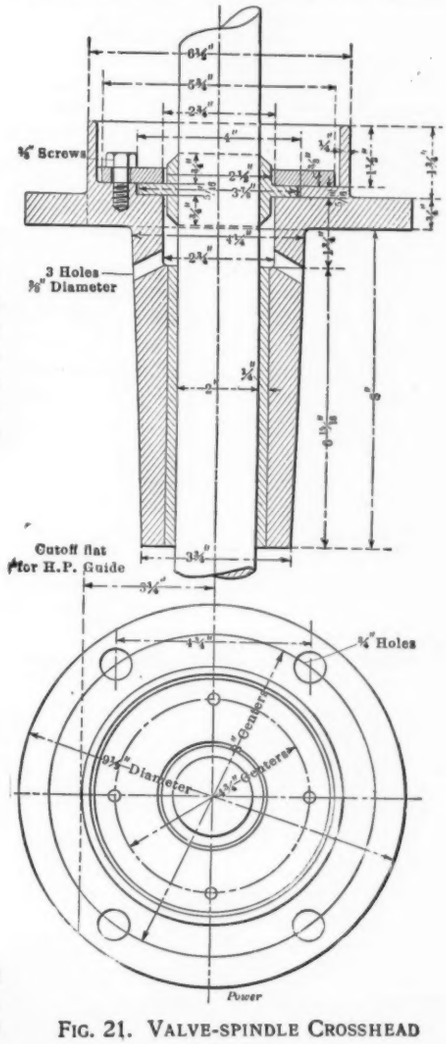


FIG. 21. VALVE-SPINDLE CROSSHEAD GUIDE

rods. A detail of this gland is shown in Fig. 18. This gland is very necessary on the low-pressure side; otherwise the crank-chamber oil would be drawn by the vacuum into the cylinder, through the stuffing box. To prevent this as much as possible, it is usual to make the distance between the low-pressure stuffing-box gland and the scraper gland at the top of the frame at least one inch greater than the stroke. By this means, the portion of the piston rod working in the cylinder never enters the scraper gland. Doors of ample size are cast in the distance piece to give access to the two glands just referred to.

In Fig. 17 is shown the low-pressure crosshead guide, distance piece and bottom-cover casting. In this case the exhaust branch is a separate casting, and is shown in detail in Fig. 19. Owing to the larger diameter of cylinder, this arrangement is rendered possible by making the cover portion slightly less in diameter than the bore of the cylinder. Soft-packing stuffing boxes are generally used for both the low-pressure piston rod and the valve spindle.

The receiver between the cylinders is detailed in Fig. 20. This consists of a plain pipe making as short and as straight a connection between the cylinders as possible. The pipe is lagged with nonconducting material, and is covered with planished-steel sheets. All pockets where it would be possible for condensed steam to lodge are avoided and the only place where water could collect between the cylinders—the bottom bend in the receiver pipe—is drained, usually being connected to a steam trap. Reheaters for receivers have been found useless and are never fitted.

In Fig. 21 is shown the valve-spindle crosshead guide. This consists of a cast-iron bracket fitted with a brass bush. As the spindle is lubricated by oil under pressure, two holes are drilled just above the bush to allow the oil, as it escapes past the spindle, to flow back again into the engine base. A brass bush which acts as a scraper is fitted at the top of the guide to prevent oil passing up the spindle, and at the same time to prevent water from the gland getting into the crank chamber. The valve-spindle crosshead guides are alike for both high- and low-pressure cylinders.

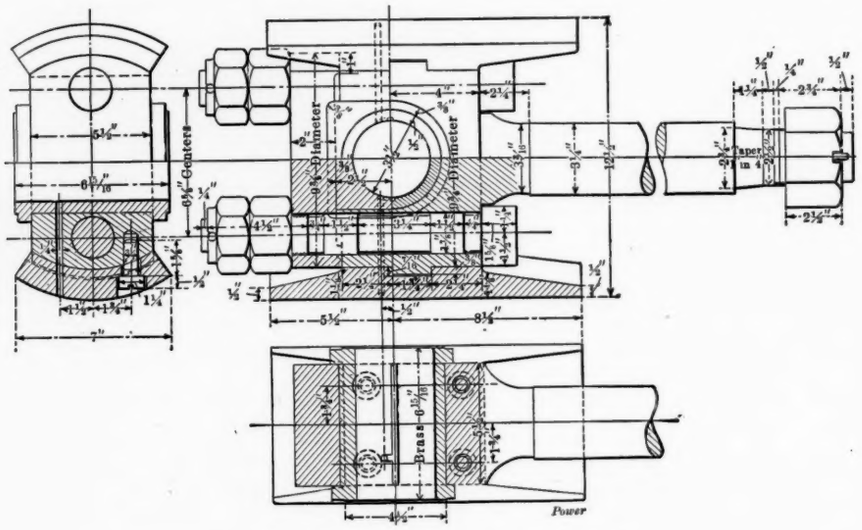


FIG. 22. PISTON ROD AND CROSSHEAD

PISTON ROD AND CROSSHEAD

Fig. 22 shows in detail the piston rods and crossheads. These are usually forged in one piece, for engines up to 500 horsepower, but for larger sizes the rod is made separate from the crosshead, and a double-ended connecting rod adopted.

The piston rod and crosshead are made of Siemens Martin acid steel and the slippers of cast iron. The latter are secured to the crosshead by cheese-headed screws made of delta metal or brass; the latter metal saves the guides in case a screw should come loose. The usual method of locking these screws is to cut out a small notch in the slipper near the head, and afterward rivet over the screw head into this notch. As adjustment is rarely necessary during the life of the engine, owing to the efficient system of lubrication, this method of locking the screws answers very well. To prevent any shearing strain coming on the screws, a feather is formed on the crosshead, which fits into a recess in the slippers.

The bushes are usually made of gun metal or phosphor bronze, and adjustment is provided by means of a cap and bolts. It will be seen that a lip is provided at each end of the cap plate to butt against the crosshead. This has been found necessary in order to prevent distortion of the crosshead when tightening up the bolts. Liners are rarely fitted to the bushes, and the nuts are usually hammered home solid, to prevent the possibility of their slacking back due to vibration when running. In addition, it is usual to fit an ordinary nut and a locknut, the latter being of the "castle" type.

The crosshead slippers are not quite central with the pin, this departure from ordinary practice being necessary to make the crosshead nuts accessible with a spanner. For attachment to the piston, a conical end is provided, in addition a good shoulder, and the nut is secured by a substantial cotter. The stress allowed at the bottom of the thread on the piston nut rarely exceeds 7000 pounds per square inch, and on the crosshead bolts

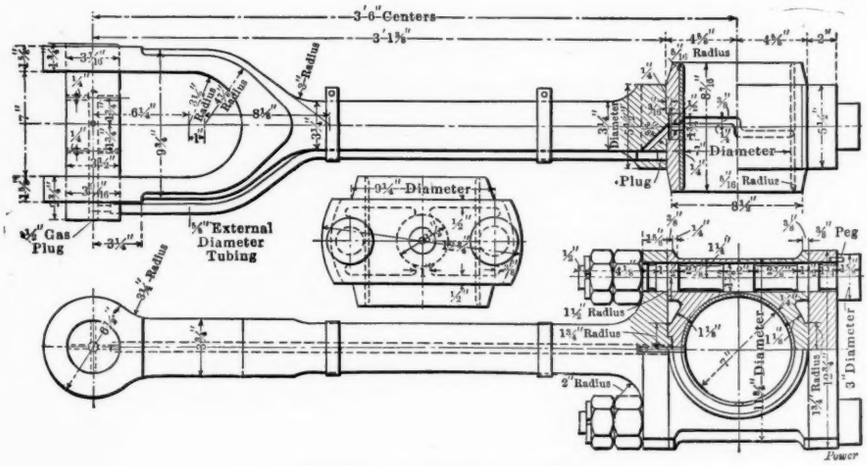


FIG. 23. CONNECTING ROD

6000 pounds per square inch. As a good shoulder is always prov beyond the cone, to prevent the piston being jammed on the rod, the stress on the body of the rod usually does not exceed 3200 pounds per square inch, and as the rods are short, owing to the shortness of the stroke, they are very stiff.

The crosshead slippers are made of ample proportions, a pressure of 40 pounds per square inch being rarely exceeded. The same remark applies to the crosshead pins on which a pressure of

900 to 1000 pounds is the standard practice. The crosshead is made very compact, with a view to keeping the diameter of the guide small, and reducing the weight to a minimum. A very strong and solid crosshead is the result.

Connecting Rods

The connecting rods are detailed in Fig. 23. These are made of Siemens

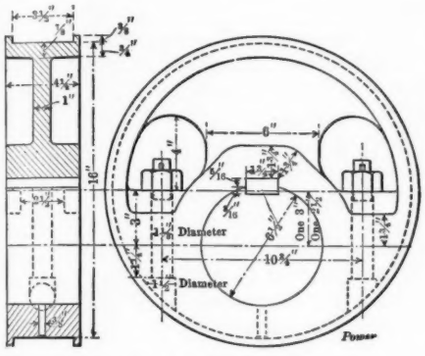


Fig. 24. Eccentri

Martin acid steel and are of the ordinary "marine" type. The crosshead pins are made of low moor iron or mild steel low in carbon, having a tensile strength of from 48,000 to 56,000 pounds per square inch, and case-hardened and ground. The most usual method of securing the pins to the rods is by shrinking, but a few makers force them in by hydraulic pressure. Shrinking is preferable, however, as it makes a tighter job owing to the small section of metal in

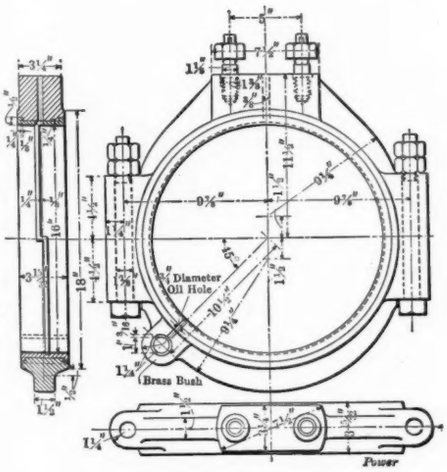


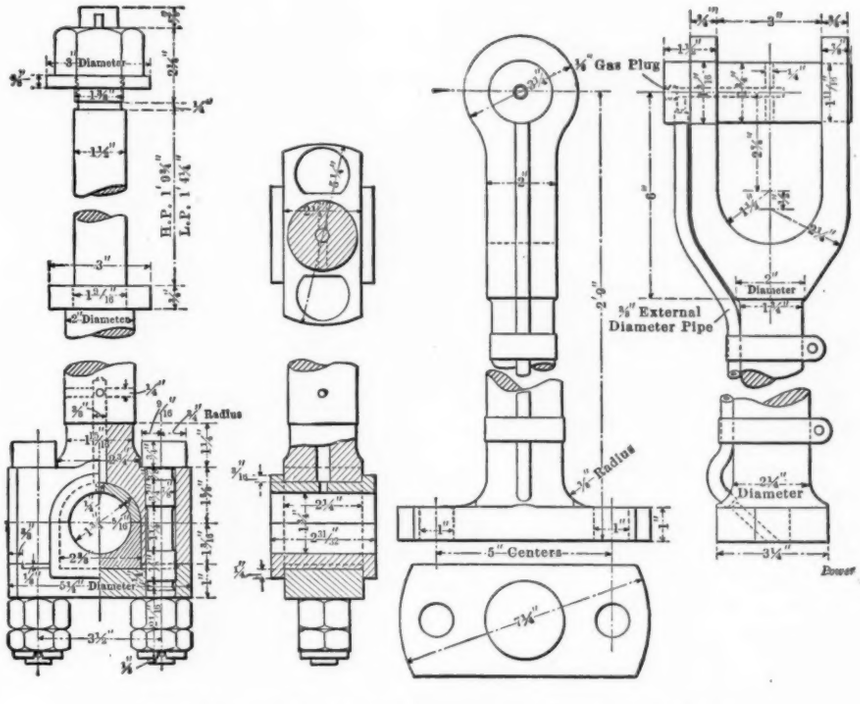
Fig. 25. Eccentric Clips

the jaw. The crank-pin bushes are usually made of brass, and lined with antifriction metal. For large engines, cast-steel bushes are often used, and a few makers use cast steel for all sizes. The method of locking the bolts is the same as for the piston rod. The crank pins are made of liberal proportions, the pressure per square inch not exceeding 400 pounds under ordinary working conditions.

The crank pin is lubricated by a hole drilled through the crank shaft, which connects with a groove in the main bearings, and a similar groove is cut around the crank-pin brasses. The groove is staggered so as to cause equal wear over the crank pin. To lubricate the crosshead pin, a pipe is led up the side of the rod as shown, and soldered into the rod and pin at the respective ends. In addition, two clips are fitted round the pipe, to prevent it from vibrating and working loose at the joints. It will be noticed that the crosshead slides are lubricated from the pin through a central hole, which communicates with a groove in the crosshead bushes; and by means of an additional hole through the crosshead and slipper, the oil is led onto the guides.

Valve Gear

The valve gear illustrated in Figs. 24 to 27 is of the simple eccentric type, but of generous proportions. The eccentrics shown in Fig. 24 are of cast iron, in halves and bolted together, no cotters being used throughout the engine except as a locking device for nuts. They are made very wide to provide a large bearing surface, and are lubricated by oil



Figs. 26 and 27. Valve Spindle and Connecting Rods

under pressure from the nearest main bearing, by a hole drilled through the crank shaft. The eccentric clips, Fig. 25, are also made of cast iron, and lined with antifriction metal. An oil groove is cut inside the eccentric clip to distribute the oil around the eccentric, this groove, which is of the staggered form, being always in communication with the oil hole in the eccentric. For locking purposes, the eccentric-clip bolts are provided with lock nuts and split pins, while the studs for securing the eccentric rod to the clips are prevented from slacking by means of a collar.

The valve spindle and eccentric rods are shown in Figs. 26 and 27, respectively. These are of stiff proportions and may appear to some to be excessively strong. It may also be thought that such heavy parts are liable to cause trouble by vibration or by their inertia effect, but on the contrary they do not give any trouble, and weak gears are liable to give out and break down. The valve is securely gripped between the nut and washer on the spindle. Below the washer, the spindle is increased to 2 inches diameter and is thus large enough to form a guide. The crosshead end is of precisely the same construction as the piston rods, being fitted with gun-metal or phosphor-bronze bushes, made adjustable by a steel cap and two bolts.

The eccentric rod, Fig. 27, is fitted with a case-hardened pin at the crosshead end, and the lubrication is arranged in a similar manner to the main connecting rod. The crosshead guide, however, is lubricated by a hole drilled up the spindle from the crosshead end, two small holes drilled from the outside leading the oil onto the guide.

The valve spindles are usually made of material similar to the piston rods, and the eccentric rods are made of material similar to the connecting rods. The reason why a different steel is used for the piston and valve rods is to provide a harder material for working through the stuffing boxes, etc. If soft steel were used, it would wear rapidly.

Noiseless Circulation of Water

By William Kavanagh

In many hotels and apartment buildings the noise of the water pump is extremely annoying. The pneumatic system of circulation illustrated herewith has been used successfully to eliminate the noise.

The circulation of water in large hotels and apartment buildings is receiving considerable attention from engineers and others connected with their erection and equipment.

In the old method a tank was placed on the roof of the building, the water being pumped up to it, from which it circulated by gravity to various points of outlet. Under this system an accidental lowering of the water in the roof tank below the outlet pipe would cut off the circulation of water to every point.

Air Circulating System

The system illustrated herewith may be found in operation in some first-class houses. With it, it is almost impossible to be without at least some water, in some part of the building. Under the old system of water circulation the noise of the pump destroyed the comfort of those in the building. Under the pneumatic system of water circulation the noise of the pump is not transmitted along the water lines.

The illustration shows the entire plant in duplicate for a building requiring the noiseless circulation of water. The erection and operation will be easily understood by noting the illustration. The two tanks *A A* are of a size sufficient to supply a building of 300 rooms. In this case each tank is 10 feet high and 5 feet in diameter, built out of boiler iron and made to withstand whatever pressure is necessary to force the water to the highest point. The pumps *P P* take filtered water from the filtering tanks *F F F*, and discharge it into the pressure tanks *A A* through the connection *D*. Pipes *C C* extend down to within 2 feet of the bottom of the tanks.

The Westinghouse air compressors, shown at *E E*, are connected to each tank, the governors of the pumps and compressors being controlled similarly to the method used with hydraulic elevators. Pressure gages are attached to each tank, as shown at *G*.

Operation

The system being without water or pressure, the pump governors will be wide open and the pumps, when started, will fill the tanks to a hight indicated by the dotted line at *M*. When the air compressors are started, the air pressure in the tanks *T T* will force the water up through the pipes *C C* to the manifold *H*, to which are connected the various pipe lines throughout the building.

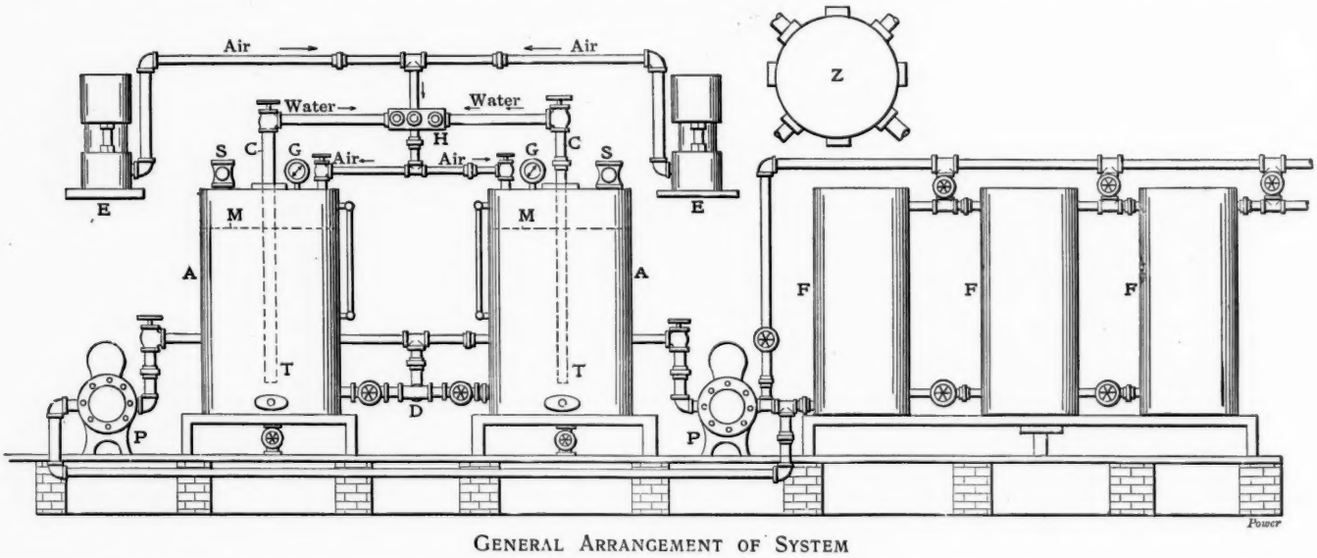
The amount of air pressure on the tanks is determined from the hight of the buildings, plus the frictional resistance offered by the water in the pipes; the governors are set slightly above this pressure, which keeps the pumps in motion and free from condensation, thus enabling the pumps to start the moment water is drawn in any part of the house.

Should both pumps fail, the air compressors would continue to circulate the water until the water had lowered sufficiently to expose the ends of the pipes *C C*. Should the air compressors fail, the pumps would continue to force filtered water through the system, but accompanied by the old objectionable noises. As the tanks, etc., are in duplicate there is but little danger of a failure. This duplicate arrangement enables the pumps and air compressors to be packed and repaired; the tanks can also be cleaned without disturbing the water circulation in the building.

The plan view *Z* is of the header *H*, showing how independent lines can be taken from it. Each line is fitted with a valve and can be shut off and repaired independently of any other line. Manholes are provided in each tank to enable them to be cleaned out, and a bypass water connection is usually made around the filters, to be used whenever the filters need cleaning.

In some plants the filters are not bypassed, but a large open or closed tank, capable of storing one or more days' supply of filtered water, is employed, and, instead of the pumps drawing water from the filters direct, they are supplied from the storage tank. This is undoubtedly the best plan, because it affords a slow filtration of the water, and the extra supply is a very commendable feature, since the mains supplying water to the building might fail. In fact, a large number of important buildings are now equipped in this manner, a supply of water being retained in tanks, which, in case of emergency, can be used when the supply tank is used in conjunction with the filters. They are elevated above the storage tank, a float being used to determine the amount of storage.

The safety valves *S S* are attached to each pressure tank, and should an overpressure occur, the safeties will discharge the surplus water back into the storage tank; thus the system becomes almost automatic in its mode of operation.



General Arrangement of System

Electrical Department

Primer of Electricity

By Cecil P. Poole

Especially conducted to be of interest and service to the men in charge of the electrical equipment.

Dynamo Armature Cores

The diagrams of armature windings given in previous lessons served to illustrate the principles of arranging and connecting the conductors in armature windings. They were not intended to show the way in which the conductors are placed, mechanically, on the armature core.

In all early dynamos, the armature cores were made up of smooth disks or rings, as in Figs. 60 and 61, and the wire was wound on the surface of the core, as indicated by the heavy black lines, except that there were many turns of wire in each coil and the entire surface of the core was covered with wires. Consequently, cross-sections of these cores would look about like Figs. 62 and 63.

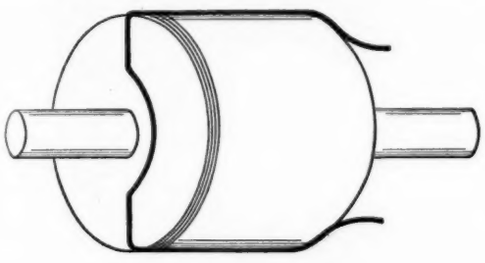


Fig. 60

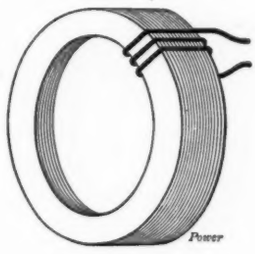


Fig. 61

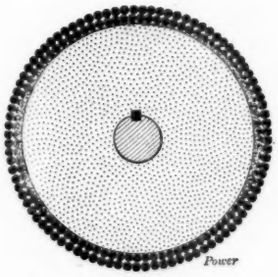


Fig. 62

The ring-shaped core was abandoned a good many years ago partly because a much greater quantity of wire is required to generate the same electromotive force than is required on a drum armature, but chiefly because of the great difficulty of putting the wire on the core. Fig. 61 shows that the wire must be threaded through the interior of the ring as many times as there are convolutions in the winding. There are other disadvantages in the ring winding, but the two mentioned were ample to put this type of winding out of use. The all surface or drum type is the only one made today, and for this reason only that type will be referred to throughout the remainder of these lessons.

Torque

A serious disadvantage of the surface-wound drum armature is the tendency of the winding to slip bodily around the core. This is due to the magnetic pull on the conductors. When a wire is moved across a magnetic field to induce an electromotive force in it, a definite force must be applied to it in order to move it through the flux, if the ends are connected together so that a current can flow. If this were not so, electrical energy could be generated with practically no application of mechanical energy, which would violate one of the fundamental and absolutely rigid laws of physics.

Now, if an effort is required to move a wire through a magnetic field in developing electrical energy, then the magnetism must pull on the wire, and that is exactly what happens. Therefore, when an armature is revolved between magnet poles, if the circuit connected to the winding is closed, there will be a backward mechanical drag on each wire which is directly proportional to the current in the wire and the flux in the magnetic field. The approximate relation between the different quantities is

$$\frac{\mathbf{B}\,l\,i}{11{,}302{,}360} = Lb.$$

in which **B** represents the magnetic density of the field in lines per square inch of field cross-section; l is the length of that part of the wire which actually passes through the field; i = the current in the wire, and $Lb.$ = the pull on the wire in pounds.

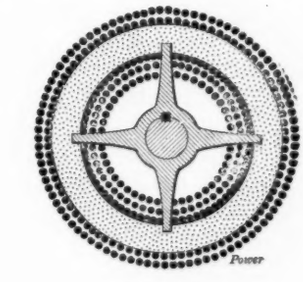


Fig. 63

The force that a complete winding has to withstand is much greater than you would be likely to think it. For example, if an armature had 300 conductors around the surface, each wire 10 inches long and carrying 100 amperes, and if the armature revolved in a field of 50,000 magnetic lines per square inch, the average pull on each wire would be

$$\frac{50{,}000 \times 10 \times 100}{11{,}302{,}360} = 4.42$$

pounds, so that the pull on the whole winding would be

$$4.42 \times 300 = 1326$$

pounds. It is easy to understand, therefore, that there would be great difficulty in holding the winding on the core of a large machine.

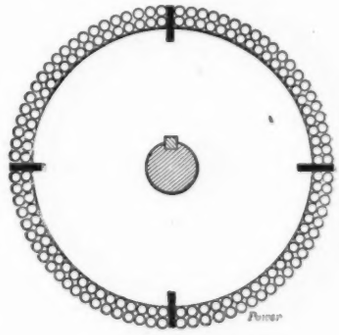


Fig. 64

To make sure that the windings of large armatures would not be dragged back around the cores, it became necessary to cut slots in the cores and set driving strips of hard wood or other strong insulating material in these slots, as represented in Fig. 64; then the winding *had* to go with the core. The winding was bound to the core surface by hard brass wires wound over strips of insulating material, as shown in Fig. 65.

Although driving strips, as shown in Fig. 64, will prevent slippage of the winding, the surface-wound armature has the disadvantage of compelling the use of a very long airgap between the polefaces

and the armature core. If the winding is ½ inch thick on the surface of the core, the airgap must be slightly more than ½ inch long to prevent the poleFaces from scraping the wires. This entails the use of an excessive amount of magnetizing force to drive the flux across the airgap, and a large magnetizing force

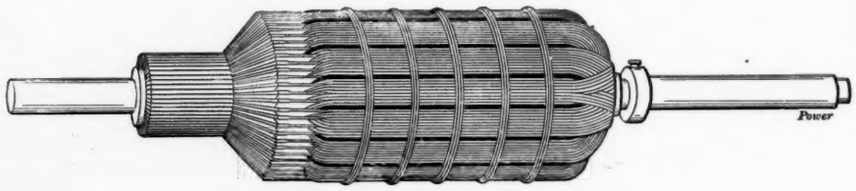


FIG. 65

requires either a large magnetizing current or a large quantity of wire in the winding which produces the magnetic flux. This will be discussed in the next lesson, when the source of the field magnetism will be taken up.

THE SLOTTED CORE

Because of the disadvantages mentioned, it is the universal custom now to

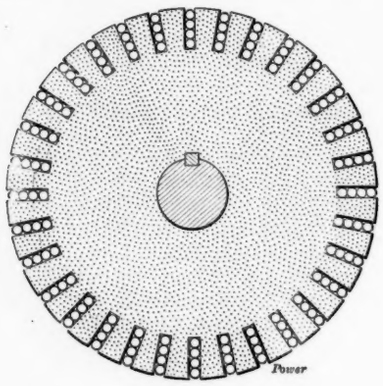


FIG. 66

put the armature winding in slots cut in the core much like those in which the driving strips were set on surface-wound armatures. Fig. 66 illustrates the commonest form of slotted armature core; it represents a cross-section through the core, showing the cut-off ends of the wires in the slots.

This arrangement permits the airgap between the poleFaces and the ends of the armature-core teeth to be reduced to the small distance necessary for mechanical clearance. It also protects the winding from accidental mechanical injury and relieves it of a great deal—nearly all—of the magnetic pull, which is almost entirely taken by the armature teeth.

It entails the slight disadvantage that the magnetic flux passing from a poleFace to the armature is not smoothly distributed over the poleFace, but is divided into as many concentrated "bunches" as there are armature teeth under a poleFace, as represented roughly in Fig. 67.

It might seem that when the wires are sunk in slots, and the magnetic flux passes from the poleFace to the armature core through the teeth, the wires will not cut the magnetic lines and will therefore not generate any electromotive force. This is, of course, not true; if it were, machines would not be built that way.

It is, furthermore, a fact that the wires generate exactly the same e.m.f. in the slots that they would on the surface of a smooth core with the same magnetic flux passing between the magnet poles and the armature core. That is, if the flux passing through the teeth to the core body of the slotted armature is the same as that passing across the airgap of a smooth-core machine, each wire in the slots will generate the same e.m.f. as each wire on the surface of the smooth-core armature, if the speeds are the same in both cases.

The reason for this is that the wires in the slots cut across the magnetic flux in the teeth, because as the armature revolves each "bunch" of flux shifts from tooth to tooth progressively. This shifting and flux cutting may be explained somewhat crudely as follows:

In Fig. 67 the group *F*, of magnetic lines, is represented as passing from the poleFace edge to the armature tooth *m*, between the slots *f* and *g*. As the armature rotates, this particular "bunch" of flux shifts back to the next tooth *n*, between the slots *e* and *f*, and when that tooth reaches the position previously occupied by the tooth *m* (see Fig. 68), all of the lines that previously passed to that tooth will pass to the tooth *n*, as represented in the sketch.

In shifting from the tooth *m* to the tooth *n*, the flux *F* necessarily cuts across the wires in the slot *f* and induces an electromotive force in them. (It makes no difference whether one considers the flux as cutting across the wires or the wires as cutting across the flux. It is relative displacement of the one with respect to the other that generates the electromotive force.)

The shifting process is carried on also by the "bunches" of magnetic flux in the other core teeth. When the armature rotates exactly the distance of one slot pitch—that is, the distance from center to center of two slots next to each other—from the position shown in Fig. 67 to that of Fig. 68, the wires in the slot *a* will cut the flux *A;* those in *b* will cut the flux *B*, and so on, in all of the slots within the reach of the magnetic flux under the poleFace.*

As dynamos are usually built, practically all of the magnetic flux that passes through the body of the armature core is carried from the airgap to the core through the teeth. Some of it does pass down through the coil slots but the proportion is so small that it is usually ignored.

A single glance at Fig. 66 will show that the magnetic path through the teeth is necessarily much smaller in cross-section than that through the core body. Consequently, the magnetic density in the teeth is much higher than that in the core. For this reason it is common practice, in estimating the hysteresis and eddy-current losses of an armature,† to consider the teeth and the body of the core separately.

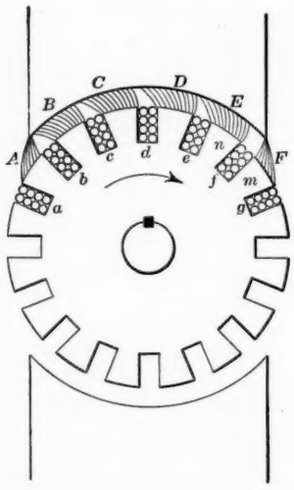


FIG. 67

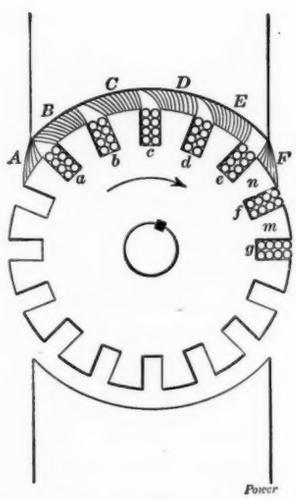


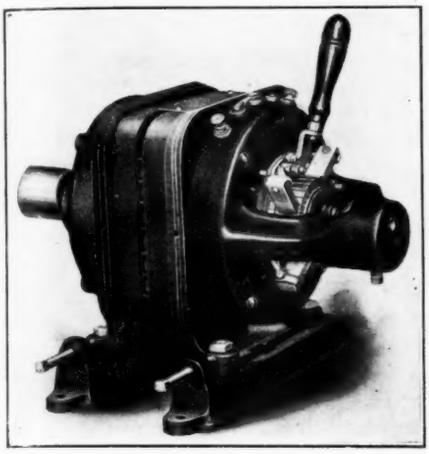
FIG. 68

*This explanation is not scientifically accurate because it is incomplete. The actions are not quite as simple as described but the results are the same as though they were and it is impossible to give a complete exposition simply.

†See pages 714, April 19, and 804, May 3, 1910.

The Kimble Alternating Current Motor

There are few small alternating-current motors that operate satisfactorily with variable speed of wide range, or even as well as the smallest variable-speed direct-current motor. The Kimble alternating-current motor is one of the exceptions to the rule, since it operates over a wide range of speeds. In construction, the single-phase motors, which are built in sizes from ⅛ to 1½ horse-



THE KIMBLE MOTOR

power, are four-pole machines, very similar to the ordinary series-wound direct-current motor, except that they are built with laminated field magnets. They operate at any rate of speed within a wide range without the use of either external resistors or induction coils—in fact, without any form of outside controlling device. With the exception of the ⅛ and ¼ horsepower sizes all of the motors are reversible.

The entire control of the motor, including starting, adjusting the speed, reversing and stopping, is vested in a single lever which merely shifts the brushes around the commutator and actuates a switch in the supply connection. When the lever is in the "off" position no current passes through the motor.

The armature is wound with wire made of a composition which is of higher resistance than copper, the object being to prevent the flow of excessive current in those coils short-circuited by the commutator brushes. The winding is said to be so proportioned that the current in the short-circuited coils cannot exceed full-load current under any possible operating conditions.

There are no "steps" in the speed gradation, because the position of the brushes on the commutator determines the speed; therefore, any rate of speed from zero to maximum may be had and the gradations are imperceptible without the use of a speed indicator. In operation the brushes always remain in contact with the commutator, the operation of the machine being almost exactly the same as that of an ordinary direct-current series-wound motor.

It is claimed that these motors start with full-load torque and that the starting current is no greater than the full-load running current. It is also stated that the motors show an unusually high power factor and good efficiency.

In the polyphase machines, which are built in sizes up to 7½ horsepower, the polar area is large, making possible a much lower no-load current than in the single-phase type. The polyphase motors are not designed for high starting torque, but rather for low slip; the starting torque is ample for most purposes, however. Tests of the 1-horsepower size are said to have shown 83 per cent. efficiency and 81 per cent. power factor at full load.

The tendency in the design of all the Kimble motors is toward short lengths and large diameters, which, of course, permits the use of short, stiff shafts and large bearings set close together. These machines are built by the Kimble Electric Company, Chicago.

LETTERS

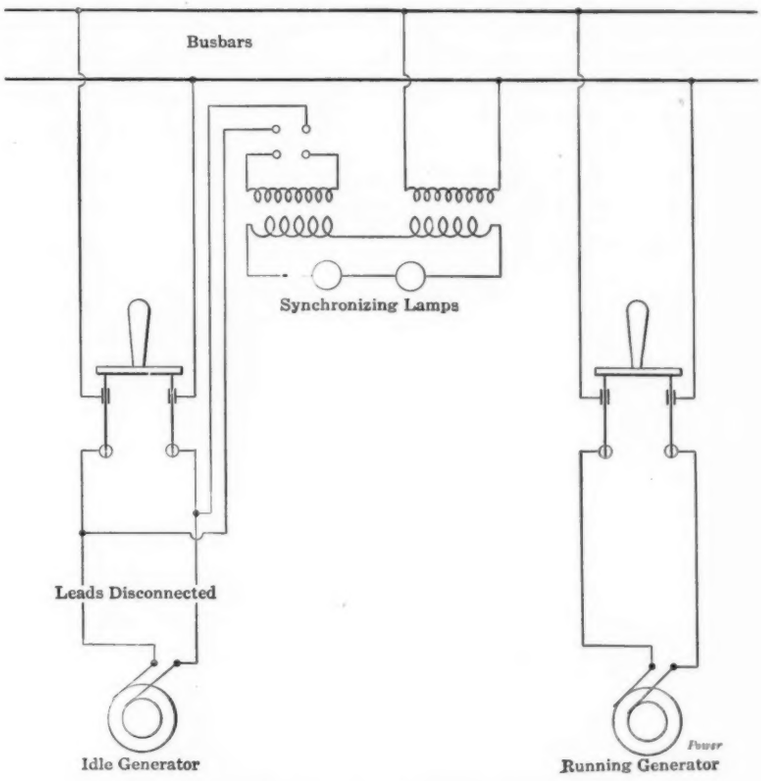
Testing Synchronizing Lamps

In a certain electric-light plant there are two alternators operated in parallel. The night engineer quit suddenly and an engineer was employed who had before operated alternators in parallel. During his first shift when the load demanded the second generator, he started it and thought he had it synchronized, but as the switch went in there were "doings." He then sent for an electrician who knew that the synchronizing lamps could be connected either for light or dark synchronizing, and he spent about an hour tracing out connections to find out which way the synchronizing transformers were connected.

In a case of this kind all that is necessary is to raise the brushes or disconnect the leads at the terminal board of one generator and throw in its switch; then run the other generator and plug in the synchronizing lamps on the disconnected leads of the dead generator. The lamps will then indicate which is right, light or dark. The diagram illustrates the principle of the method for a single-phase plant; it applies just as well to polyphase work.

F. W. REED.

Havre, Mont.



CONNECTIONS FOR TESTING SYNCHRONIZING LAMPS

It is estimated that the Panama canal when finished will have cost the United States nearly $500,000,000. Yet, of our annual exports and imports amounting in value to about $4,000,000,000, less than 8 per cent. are carried in American ships. Is it not about time that we awoke to the need of rehabilitating our merchant marine?

There are water-power possibilities within a radius of 45 miles of Ottawa, Ont., which it is estimated amount to about 900,000 horsepower.

Gas Power Department

The Necessity of Thoroughly Cleansing Producer Gas

By L. L. Brewster

It has been my privilege to visit a considerable number of gas-power plants in the past few years and in almost every one I have discovered conditions which lead me to believe that altogether too little attention is given to the cleansing of the gas before delivery to the engines, more particularly in plants using producer gas.

The negligence in some cases seems to be in the builders of the producer equipment; in others, the plant attendants are at fault, and in many cases the fault is on both sides. It would seem, in view of the disastrous experience of some of the earlier plants, that both makers and operators would be thoroughly alive to the importance of clean gas; but recent observations do not show that this is so.

Almost everybody familiar with the history of the gas engine in this country will remember the troubles due to dirty gas which were experienced in the plant of the Lackawanna Steel Company at Buffalo, N. Y. Cleansing apparatus was not then developed to the efficient condition of today, but evidence that the lesson of that experience was well learned by steel makers is afforded by the elaborate and highly efficient cleansing plant at the Gary plant of the United States Steel Corporation.

The description of this plant and the results obtained, by H. J. Freyn, in a paper presented at the spring meeting of the American Society of Mechanical Engineers, should make a deep impression on any reader. In his words, "Not so many years ago it was thought in good faith that gas engines could operate on dirty gas, and it required years of costly experimenting to develop the art of gas purification to its present perfection, after the far-reaching importance of the problem was at last understood."

The overall efficiency of the wet scrubbing and secondary cleaning plants at Gary was 99.5 per cent., which reduced the dust content per cubic foot to 0.00583 grain and yet this amount meant that 3.3 tons of dust entered the engines during the year 1909.

Accurate figures showing the results of gas cleansing in producer plants are difficult to obtain, principally because accurate determinations are not regularly made and kept, but the effect on the engines speaks for itself in many plants I have seen.

Everything worth while in the gas engine and producer industry will be treated here in a way that can be of use to practical men.

Some Results of Inadequate Cleaning

I recall very well the experience at a plant near New York in which is a 300-horsepower double-acting tandem engine using gas from an anthracite suction producer. After only two months' run the handhole plates on the fuel duct leading to the inlet valves were removed and dust by the handful was brushed out. So much dust had entered and evidently settled on top of the piston rod that the wear was perceptible. Large particles of dust and much fine dust had so worked into the stuffing boxes as to fill all the grooves and put the packing rings out of business. This caused the box to leak and soon the inside of the bed forward of the crank-end cylinder was smeared with a dirty mixture of dust, oil and water. When the boxes were removed it was found necessary to drive some of the packing rings out of their grooves with cold chisels, so tightly had they been wedged about with a mixture of dust and lubricating oil.

Now, it will probably be said that the scrubbing part of this plant was entirely inadequate, but it was fully up to the then best practice both as to capacity and arrangement. The producer had an internal diameter of 6 feet 6 inches; the wet scrubber was 4 feet 6 inches in diameter and 20 feet high; the dry scrubber was built with two compartments, one above the other, and each was 8 feet in diameter and 4 feet 3 inches high.

In my opinion the spray pipes in the wet scrubber were not well designed. They were simply a pair of pipes placed at right angles and perforated for their entire length with small holes. The distribution of water in the scrubber was probably faulty.

I visited this plant recently and found many evidences that dirty gas has been flowing to the engine in the intervening three years. A set of piston rods have been replaced and the old ones show wear right up to the limit of the packing-ring travel. The surface of the worn part has the appearance of having been lapped, it is so smooth.

I believe this excessive wear was due largely to the presence of fine dust, which with the lubricating oil had the effect noted.

I found also that the suction at the fuel duct in the engine had increased from 4 to 20 inches, water pressure, and on examination found that the elbows in the gas pipe were packed so full of dust that there was only a small opening next the inner radius for the gas to pass through.

In another plant, also near New York, there are two anthracite suction producers and two three-cylinder vertical engines, controlled by governors which throttle the mixture. It is necessary to remove the throttle valve of each engine every second day and clean it to prevent it from sticking and becoming inoperative.

The producer equipment in this plant is that of a well known maker and the water pressure at the scrubbers is 30 pounds per square inch. In fact, the conditions are those asked for by the makers and the attendance is of the best. Yet these engines are expected to run on gas of the dirty character indicated, and do, but only by frequent cleaning.

In another plant which I see occassionally, a 150-horsepower horizontal double-acting engine is supplied with gas from an anthracite suction producer of equal rating. The throttle valve first installed with this engine was a ported sleeve rotating about a vertical axis. It was found soon after starting the plant that it was necessary to remove and clean this valve every day in order to prevent it from sticking. The makers of this engine have found that the piston type of throttle valve is simply put out of business by dirty gas in so many plants that they have replaced this type with one having balanced double disks which come down on two seats and have no sliding surfaces. It does not seem fair that builders of engines should be compelled to furnish engines capable of running on dirty gas, but the makers just mentioned actually abandoned a well designed mixing and governing valve because it was almost always required to handle large quantities of dust in operation.

I might mention many other plants in which dirty gas is a constant source of trouble and cost, but enough has been said, probably, to show that such conditions exist, and I believe they exist in a majority of the producer plants now in operation.

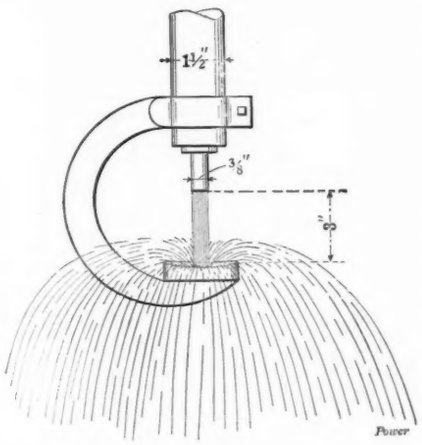
Rotary Scrubbers Necessary

It is a really difficult matter to thoroughly cleanse producer gas. A large part of

the dust is taken out in the wet and dry scrubbers, especially if the latter is partly filled with oily waste, but a certain amount of fine flocculent dust seems to pass through both scrubbers and find its way to the engine. This is true more often when a white-ash coal is used in the producer.

Mechanical scrubbers seem to be the only ones that will remove the dust to a sufficient degree and these are seldom installed in the producer plant. It is the rotary washers that are the salvation of the Gary plant, although a most efficient "static" cleaning plant is in use.

The addition of some form of mechanical scrubber would increase the first cost of a producer plant and the cost of operation, and thereby make the selling of a plant more difficult in competition with the simpler form of plant now generally offered. It would therefore be much better if, by common consent, all builders of producer plants would provide adequate scrubbing apparatus, for in the



SUGGESTION FOR A SCRUBBER SPRAY

last analysis the faulty operation must come back to them.

The theory of gas cleansing is well understood, but seems to be poorly applied. It is, briefly, that if each particle of dust be wet it will become sufficiently heavy to deposit on some surface whence it can be washed off instead of being carried along in the stream of gas. The natural process then would be either to divide the gas into a large number of small streams and pass it through a tortuous passage whose sides are wet, as in the common wet scrubber, or to pass the gas through an extremely fine water spray, or a combination of both.

A Simple Sprayer

In the ordinary wet scrubber, the water distribution is not uniform or the supply is insufficient, or both. In many cases spray nozzles become clogged, with the result that the water runs down over part of the coke only and uncleaned gas passes through the other part.

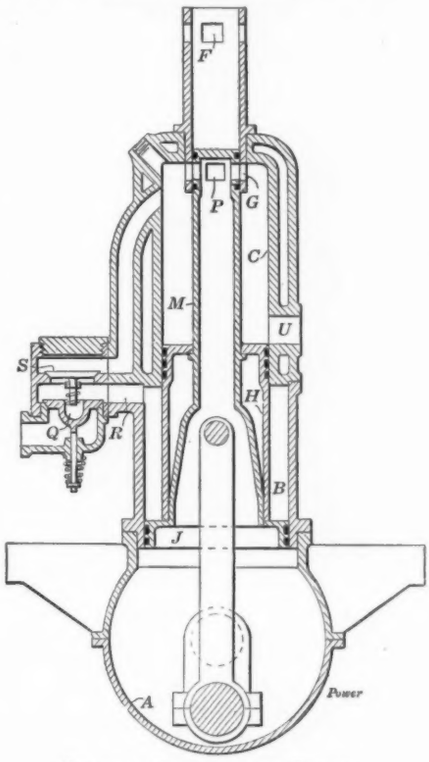
There are many forms of spray nozzle, but one which has the advantages of not becoming clogged, of distributing the water well and being easily made is shown in the accompanying sketch.

The spray pipe is reduced by a bushing to a nozzle of about ⅜ inch diameter, which should be made of the correct shape to give a clean, solid stream of water if the drop is to be considerable, but this usually is not necessary. The stream is delivered into a shallow saucer which is supported from the delivery pipe above in any convenient manner; the bracket shown is simple and easily made.

Pradeau's Two-Stroke Engine

The accompanying engraving illustrates Carlo W. Pradeau's idea of a scavenging two-stroke-cycle gas engine. The general arrangement is fairly well presented by the drawing. It includes a closed crank chamber, for the purpose of compressing air to be utilized for scavenging and a tubular extension of the piston, which works in a corresponding extension of the cylinder; this latter extension is provided with ports which communicate at one end with the explosion chamber and at the other end with the atmosphere.

The action of the engine is as follows: In the position shown in the drawing the annular chamber *B* in the lower end of the cylinder is full of explosive mixture which has been drawn in during the downstroke through the port *R* and valve *Q*. When the piston rises, the explosive mixture contained within the annular space *B* is forced out through the passage *R* and past the valve *S*, the spring controlling which is sufficiently powerful to prevent the valve from opening until the exhaust passage *U* is covered by the piston *H*, and the ports *G* and *P* are no longer in alinement. The explosive mixture is therefore forced into the top of the cylinder. The ports *P*, at the end of the upstroke, coincide with the holes *F*, allowing free ingress of air down the extension *M* to the crank chamber. The volume occupied by air at atmospheric pressure is that of the interior of the tubular extension *M*, that portion of the interior of the piston in communication with the extension, the interior of the piston extension *J*, and the crank chamber *A*. Upon the explosion the piston descends, and the air contained in the spaces just mentioned is compressed. When the top of the piston *H* passes the edge of the exhaust port *U*, the exhaust gases commence to escape through this port, and just after such opening the ports *P* in the extension *M* communicate with the ports *G*, allowing the compressed air to rush from the crank case into the cylinder *C* and drive out the burned gases. This occurs when the piston is in the position here shown. While the piston was descending, a fresh charge of mixture was being drawn in through the valve *Q* by the suction of the piston *J*, and the cycle of events is repeated.



PRADEAU TWO-STROKE ENGINE

We are indebted to *The Mechanical Engineer*, Manchester, Eng., for the illustration and the foregoing particulars.

Transatlantic Steamers with Diesel Engines

Emil L. Boas, general manager and resident director at New York of the Hamburg-American Line, has just returned from Europe full of confidence in the two freight ships which that line is building and equipping with internal-combustion engines for service between New York and Hamburg. The vessels are of 8000 tons displacement each and each will have two 1500-horsepower engines, giving them a speed of about 12½ knots. This will be the first time that this form of propulsion has been applied to vessels of such size. The engines are being built by the Augsburg & Nurnburg Company and are of the Diesel type, using the heavier grades of fuel oil. The oil can be carried in the double bottoms of the ships, doing away with coal bunkers and boiler rooms. There is thus effected great saving in space and weight, and the vessel is made more efficient as a cargo carrier. The engine-room force will be materially reduced. There will be no stokers and few oilers, as the bearings will be lubricated automatically, the oil being filtered and used continuously. It is expected that the vessels will be ready in 1912, and, if successful, will be followed by a passenger ship, propelled by the same method.

Readers with Something to Say

Practical information from the man on the job. A letter good enough to print here will be paid for. Ideas, not mere words, wanted.

A Peculiar Piston Break

Early this spring I took a position as operator on an Austin excavator. The engine was of the double-crank type with cylinders 8x12 inches in size. A pinion on the crank shaft acted directly on the masterwheels of two drums on the machine. The soil was very hard, and the builder's expert had been with the machine for eight months because he could not secure an operator who could handle it successfully on account of the hard soil. The operator who came from the shops with him used the engine very roughly. The piston rod screwed into the crosshead and was secured by a lock nut on the rod against the crosshead. This nut would come loose and the piston rod would unscrew from the crosshead. Time was never taken to adjust the piston rod properly in the crosshead, or to secure marks by which the proper adjustment could be insured without removing the cylinder head, but a good or bad guess was made and the lock nut tightened. If the engine turned over without the piston striking either end of

SECTIONAL VIEW SHOWING BREAK IN PISTON

the cylinder, all was considered well enough and the engine was run in this manner.

When I took charge the expert explained that none of the former operators were "there with the goods," that he was not there for his health, etc., and that if I did not in a few days show my ability to run the machine, I would be requested to leave the job.

I made up my mind to give the machine a thorough inspection so as to be sure that everything was in good order, before I tried to make any showing. To my surprise, however, the expert stopped me and said that the machine was in good order, and to get a move on and start the machine. I objected, saying that any good engineer always acquainted himself with his machine before he attempted any operating. He insisted that he was somewhat of a mechanic himself and that he had the running of things and did not care to take any objections from me. I was to take his word for the condition of the machine. That, of course, took the responsibility from me. I knew where my levers and the throttle were, and I started.

After a short run the piston unscrewed from the crosshead again, and when I proposed getting the proper clearance and a few marks to guide myself by should the piston unscrew again, the expert told me to "just tighten her up and let her go at that." I did. The piston rod held for a week or so, and things moved nicely. All of a sudden, the engine gave a hard jolt and nearly stopped. I shut the steam off immediately. When I tried to turn the engine over by hand I found that it would not pass the center. I took the cylinder head off and found that the piston was broken as shown in the figure. The packing ring was broken into a large number of pieces of from ½ to 3 inches in length. The rod had unscrewed from the crosshead enough to allow the packing groove in the piston to pass well over the counterbore of the cylinder. One of the small pieces of the broken packing ring dropped into the counterbored part of the cylinder and, of course, something had to go when the piston started forward. The back wall of the piston gave way as shown in the figure.

LOUIS T. WATRY.

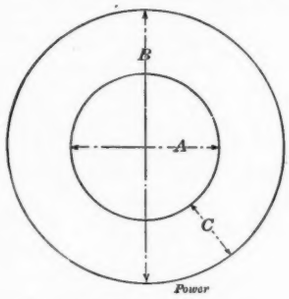
Pueblo, Colo.

Practical Packing Record

I have found the accompanying form of record very useful in operation, as well as convenient in ordering packing for the engine room.

A is the diameter of the rod.
B is the diameter of the box
C is the size of the packing.
D is the depth of the box.

It is tacked on the inside of the packing cupboard door and the cupboard is divided



KEY TO MEASUREMENTS

into compartments corresponding to the headings on the record. Thus much time is saved by not having to measure the packing and glands each time any new packing is installed.

EDWARD T. BINNS.

Philadelphia, Penn.

A Steam Heating Kink

Most engineers have experienced trouble in heating a certain part of the building when the steam would not circulate where it was required. When such trouble is encountered it is well to trace out the return pipe when the system is in use. It will probably be found that the pipe becomes cold at a certain point. This indicates an air pocket.

With a breast drill and an 8-inch bit bore a small hole at the top part of the pipe. This allows the air to escape and the return water goes on its course. In some cases it may be advisable to screw a ¼-inch petcock in the pipe so that it can be closed when the air has escaped.

WILLIAM WATT.

Lambton Mills, Ont.

PACKING RECORD

	Packing for.....	A.	B.	C.	D.	Remarks.
No. 1	Corliss engines, etc.....					
No. 2	Feed pump.....					

AVERAGE OF THE ANALYSES OF SEVERAL SAMPLES OF FUEL
COMPARED WITH THE ANALYSIS OF A COMPOSITE SAMPLE MADE UP OF THESE SAMPLES.

Fuel	Fuel as Fired															Dry Fuel.		
	Fixed Carbon Per cent.		Volatile Per Cent.		Moisture Per cent.		Moisture in Composite differing from Average		Ash Per cent.		Sulphur Per cent.		B. T. U.			B. T. U.		
	Average	Composite Sample	Average	Composite Sample	Average	Composite Sample	Per cent. of Total	Per cent. of Average Moisture	Average	Composite Sample	Average	Composite Sample	Average	Composite Sample	Difference	Average	Composite Sample	Difference
	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18
Anthracite	78.55	77.68	5.77	7.07	3.93	3.47	0.46	11.65	11.75	11.78	1.17	1.61	12,682	12,754	+ 72	13,201	13,212	+ 11
Pocahontas	73.33	73.43	19.49	19.50	2.01	1.90	0.11	5.47	5.17	5.17	0.98	1.00	14,753	14,754	+ 1	15,056	15,040	— 16
Byproduct coke	80.68	81.74	3.34	2.74	5.92	5.73	0.19	3.21	10.06	9.79	1.15	1.41	12,049	12,071	+ 22	12,807	12,805	— 2
Solvay process coke	85.80	85.76	2.14	2.45	1.60	1.51	0.09	5.63	10.46	10.28	0.65	0.65	12,504	12,490	— 14	12,707	12,681	—126
Western bituminous No. 1	42.52	42.12	39.48	40.48	9.80	8.93	0.87	8.88	8.19	8.47	3.82	3.84	11,675	11,781	+106	12,943	12,936	— 7
Western bituminous No. 2	42.53	43.74	41.38	41.49	11.14	10.09	1.05	9.42	4.96	4.68	2.39	2.37	11,990	12,139	+149	13,493	12,356	—137
Western bituminous No. 3	40.59	40.63	37.58	37.55	11.02	11.07	0.05	0.45	10.81	10.75	3.00	3.00	11,115	11,109	— 6	12,492	12,492	0
Western bituminous No. 4	41.15	41.51	37.36	38.01	10.10	8.14	1.96	19.41	11.39	12.34	2.92	3.00	11,155	11,209	+ 54	12,422	12,202	—220
Western bituminous No. 5	40.36	40.35	39.53	39.57	11.18	11.14	0.04	0.36	8.93	8.94	2.21	2.48	11,548	11,383	—165	13,002	12,810	—192
Western bituminous No. 6	40.85	41.45	36.67	36.62	10.98	10.86	0.12	1.09	11.51	11.07	3.75	3.71	10,943	10,998	+ 55	12,293	12,338	+ 45
Western bituminous No. 7	40.84	40.56	37.56	37.67	10.24	10.01	0.23	2.25	11.36	11.76	3.19	3.56	11,005	11,033	+ 28	12,260	12,260	0
Western bituminous No. 8	40.30	40.36	35.42	35.31	11.43	11.30	0.13	1.14	12.85	13.03	5.93	6.66	10,443	10,405	— 38	11,791	11,731	— 60
Western bituminous No. 9	40.06	40.48	40.22	40.37	7.94	7.65	0.29	3.65	11.78	11.50	4.58	4.55	11,331	11,412	+ 81	12,308	12,357	+ 49
Western bituminous No. 10	48.11	48.72	33.31	33.33	7.60	7.44	0.16	2.11	10.98	10.26	1.40	1.17	11,783	11,784	+ 1	12,752	12,731	— 21
Western bituminous No. 11	48.07	48.42	36.27	36.28	6.55	6.66	0.11	1.68	7.27	8.64	2.09	2.11	12,274	12,233	— 41	13,134	13,106	— 28
Western bituminous No. 12	50.85	51.00	34.93	35.02	6.57	6.20	0.37	5.63	7.65	7.78	2.00	2.22	12,642	12,602	— 40	13,531	13,435	— 96

Comparison of Coal Analyses

I recently had occasion to run a series of boiler tests in all of which the same kind of coal was used and the question arose as to the method of preparing a sample of the coal for the chemist. The choice lay between three methods: To follow the usual custom of taking a sample during each test, have it analyzed and use the results in working up the test; to average these analyses and use the resulting values in computing all the tests; or to take the usual samples, and after air-drying, mix them in the proportion of the weights of coal fired during the respective tests, then have this composite sample analyzed and use the results in computing all of the tests.

To obtain some idea of the results that might be expected from these last two methods, a study was made of analyses already available, and the results are shown in the accompanying table. This gives analyses of sixteen different fuels, and the average results are taken from six to twelve different analyses. A composite sample had already been made and analyzed for each of these sixteen fuels, and the two values for each item of the analysis are given in the table.

The analysis of the separate samples in several instances varied considerably, probably due to the manner of taking the sample rather than a difference in the coal. This may, in a measure, account for some of the uneven results obtained in working up the tests, for when they were computed by using the average values the results were very uniform.

It will be noticed from the table that with the exception of two cases the samples had lost some moisture before the composite sample was made, although this difference was small in most cases. The greatest variation in the fixed carbon among the bituminous coals was about 1¼ per cent., although in most cases it was less than ½ per cent.

The B.t.u. are given both for "fuel as fired" and for "dry fuel," and a study of the differences as shown by columns 15 and 18 is of considerable interest. In the case of fuel as fired, Western bituminous coal No. 2 showed 149 B.t.u. more per pound from the composite sample, while coal No. 5 had 165 B.t.u. less. The average of all the positive differences is 57 B.t.u. and of the negative differences 51 B.t.u. When considered on a dry fuel basis, two of the coals showed the same heat value, three a very slight increase in the composite sample and in eleven cases the heat value was less according to the composite sample. This difference reached as high as 220 B.t.u. with an average of 82 B.t.u. for the eleven samples. In the case of coals Nos. 2 and 4 this difference changed from positive to negative, a total of almost 300 B.t.u.

In view of these irregularities it is decidedly questionable as to whether it would be advisable to use the values obtained from the composite sample in computing a series of tests.

FRANK L. BUSEY.

Urbana, Ill.

Causes of Failures

It is not an unusual occurrence for an engine or a boiler to break down through causes totally unknown to the engineer in charge, and it is only by knowing how engines have failed, by carefully noting symptoms of a possible failure, and by applying the remedy before the breakdown occurs, that a man can qualify as a competent engineer. The skilled engineer's eye is trained and is quick to detect the slightest indication of trouble; also his ears are tuned to the hum of the machinery and should any unusual noise, however slight, break in upon the regular pulsations of the engine, he will at once notice it and will be able to satisfy himself as to the cause. This skill is due to mental training in conjunction with the practical experience gained through manual labor, which unfortunately is none too often appreciated by employers.

On the other hand, the unskilled engineer will run an engine month after month without inspection or overhauling, until finally a shaft breaks in consequence of a bearing being in need of adjustment, or a disregarded leaky rivet hole allows corrosion to set in; or he will

overload the safety valves of the boilers by moving the weight so as to make the boiler do more work at the risk of life and property. It is a fact that there are men in charge of engines who know practically nothing beyond how to shovel coal; yet they are not to blame for getting into the engine room, changing the shovel for the starting lever.

One cannot view with complacency the startling number of boiler explosions that occur annually in the United States, especially when it is remembered that a large part of these are avoidable, some being due to culpable negligence. The primary cause of most boiler explosions may be traced to corrosion of the plates, either as a uniform wasting of the surface, or as rapid local decline, rendering the plates too weak to withstand the ordinary working pressures. The idea of high pressure is only a relative one, and can have reference only to the comparative strength of the boiler upon which the pressure is exerted; one boiler may be as safe when carrying 200 pounds pressure as another carrying 50 pounds. Of all causes of failure, there is none so deceptive and so certain to reduce the strength of the plates as corrosion, both internal and external, especially the latter, when the sweating of the plates underneath, in conjunction with the soot and moisture, is not suspected. This dangerous condition usually does not attract the attention of the unskilled engineer, hence it does its work unarrested until ultimately the inevitable happens.

I have been engaged for over 25 years in the construction and operation of engines and boilers, and have had many opportunities of observing the causes of failure and breakdowns under the care of unskilled engineers; hence it may be of interest to cite a few cases.

In one case an engineer broke the beam of an engine into halves by allowing the injection water in the condenser to enter the cylinder; the water, being incompressible, resisted the down stroke of the piston, and the momentum of the flywheel resisted the upward tendency of the water, consequently the weakest point between the crank shaft and the condenser, which proved to be the center of the beam, parted and came down with a crash. This accident was caused soon after starting by not regulating the supply of water to the condenser so that the air pump could discharge it into the hotwell.

In another case a beam was broken by allowing blocks of wood to lie about the engine room until some were accidentally knocked into the flywheel pit, where they caught between the rim of the wheel and the masonry, forming a brake which acted against the steam in the cylinders.

An engineer allowed the lid of an oil box to fit air tight, consequently the siphon would not act properly. He retrimmed it and still the bearing continued to become hot. As his efforts were of no avail the journal and the brasses were badly cut, because the mill had to be kept running.

In another case feed water was taken from a brook and some chips were drawn through the suction pipe and into the pump, getting under the valve and disabling it. As the boiler became short of water the engine had to be stopped, and the top check valve being up, the second check could not be examined, so steam had to be blown from the boiler and the works stopped, all because of the absence of a strainer in front of the intake, and by the want of a "head" that could conceive of the chips floating in the brook.

A boiler was washed out and filled with water again and the next morning the fire was started as usual. When the man in charge began to think it time for steam to appear, he perceived something smoking on top of the boiler and upon investigation found it to be an old bag. Shortly afterward he found the flue to be red hot. It was found that the water had passed out through the blowoff cock which had not been closed. All blowoff cocks should be carefully watched, because they may fail at any time, especially when least expected, through ordinary wear and tear.

Enough has been said to establish the fact that a large percentage of failures and breakdowns are avoidable, if competent and careful engineers are in charge. When a man's mind possesses no scientific principles for his guidance, every act is invested with doubt, and any accident which may be caused through ignorance is generally surrounded with mystery.

JOHN GREEN.

Seattle, Wash.

Graphic Method of Determining Rope Sag

In the article under the above heading in the August 16 issue of POWER, the question of the sag to be allowed in rope drives is touched upon, but only an approximate rule for determining this is given. Since writing the article it has occurred to me that it would be well to supplement it with an excellent graphic method.

In getting out details of a rope drive it is necessary, for structural reasons, to anticipate fairly closely the amount to which the slack side will sag. It may, for one reason or another, be necessary to place a girder or pipe in such a position that it passes at right angles to and between the two sides of the drive, and where the lower side is the driving side there is some risk of the upper ropes coming in contact with the girder when sagging under load. Further, when the top is the driving side the floor must be kept sufficiently low to accommodate the sag of the slack side. It is not an easy matter to say offhand what the sag will be with ropes running under normal conditions, and for this reason the following graphic method will be found very useful.

The sketch illustrates the method of procedure both when the sag is at the top and at the bottom. Considering the first case, which is shown in the upper diagram, the drive is laid out to scale and two arcs are described, having for their centers the centers of the pulleys and for their radii the distance between the pulley centers and the inside periphery of

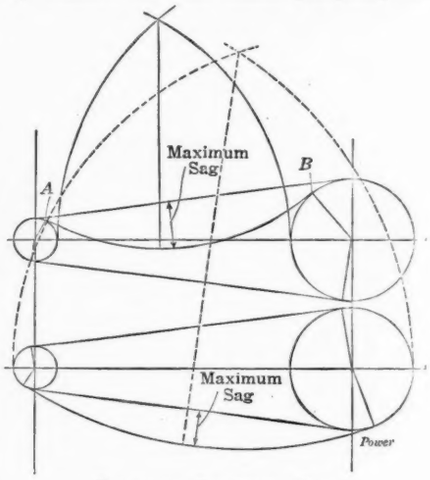


DIAGRAM SHOWING SAG

the opposite pulley at the center line. These arcs intersect at the top of the diagram, and the next step is to draw a line radially from the centers of each pulley toward this point of intersection. These lines touch the peripheries of the pulleys at certain points as A and B, and if a third arc be described, having for its center the intersection of the first two arcs and for its radius the distance between this point and either A or B, it will show correctly the maximum amount of sag to be allowed for. This may be scaled in feet and inches by measuring the distance between the point of extreme sag and a line running at a tangent to the two pulleys, as shown.

In the second case, where the slack side is at the bottom the procedure is similar to the foregoing except that the radii of the first two arcs, indicated in the lower figure by the broken lines, are the distances between the pulley centers and the outside periphery of the opposite pulley at the center line.

In practice, when the sag exceeds the amount arrived at by this method it is time the ropes were taken up. In the article previously referred to, the allowable sag is given as from 8 to 10 per cent. of the distance between the pulley centers. The result of the graphic method gives a larger sag which must therefore be taken as the maximum.

FRANCIS H. DAVIES.

London, England.

Questions Before the House

The Massachusetts License Law

As an engineer in the State of Massachusetts, I am naturally interested in all discussions dealing with its license law, and have read lately of a desire on the part of some men to make the law more strict and also to make it harder for engineers by contesting their applications for license.

In these times of close competition, when men must do more, know more and earn more than ever before, I think we should move slowly in trying to hinder a man when he is trying to better himself.

When a man is known to be unsafe, a user of intoxicants or notoriously careless, it is right that he should not be allowed to handle steam machinery. There are, however, many men, honest and conscientious in purpose, who do not have the nerve to match wits with an examiner, yet would make good and faithful workers in any plant.

I know, and I have no doubt that there are others also who know, of cases where such men have not been given a fair show, when being examined for license. and have become well nigh discouraged.

The license law was passed in order that we might have engineers and firemen who would be safe to intrust with the operation of steam boilers and machinery so that the public would be protected. When this end has been attained, it is of no consequence to the inspectors or the public how far the engineers may go into the finer points of the business, such as economy and efficiency of operation. These depend upon the men themselves and just as far as they develop their talents will they attain greater success. But I do not know that they will make any safer men than those who are not as smart mentally. If the action of the law were independent of the vagaries of human nature, we should have no complaint about one man passing the examination easier than another, when both had the same ability. But the inspectors are but men subject to the same changes of nature.

The examiners have the law on their side, and to appeal to the board would be still worse for the engineer, for three men can disconcert an applicant much more easily than one. For the examiners I will say that they are often imposed upon, and that there are some who are perfectly fair and just.

In conclusion, considering all these things and the fact that the present law is a good one and is enforced as well or better than the average, when a man has had the required experience and satisfied the examiners of his fitness for a license, he should not be hampered by any man or group of men in getting his license. As we have a department of State officials for the purpose of deciding who shall receive a license and who shall not, why not leave the question to them?

G. H. KIMBALL.

East Dedham, Mass.

Comment, criticism, suggestions and debate upon various articles, letters and editorials which have appeared in previous issues.

Installing Globe Valves

Much has been written about the proper way in which to install globe valves. I would like to add my experienc to the rest. I received my training in the engine room of a lake steamboat, where safety is the first consideration, and have worked with pressures of from 40 up to 275 pounds. Globe valves are invariably installed with the pressure on the under side of the disk. I have never known a single valve to fail by the stripping of the threads on the stem. I have taken valves apart that had been in constant service ten months in the year for 20 years with the pressure on the under side of the disk and have seen no signs of wearing or stripping of the threads. In fact, the only trouble I have ever had was with valves where the pressure was above the disk.

As E. F. Tracy remarks, in the August 30 issue, more harm is done with the monkey wrench than by wear. I never use a wrench nor allow one to be used to close a valve. If the handwheel will not close it tight it is time to unship and grind the seat or replace the disk.

I know of valves that have been in service under 275 pounds pressure for nearly ten years in which it would be impossible to detect any wear of the threads or stem. The valves are to all appearances as good as ever.

I know of an engine which has been in operation for over 25 years, working under 90 pounds pressure. The pressure is on the under side of the throttle valve. This valve has been opened and closed an average of once a day all that time, yet the valve is steam tight and has no lost motion, showing that the threads are not worn a particle.

So far as globe valves in feed-water lines are concerned, there is room for argument. I have seen valves placed both ways with satisfactory results. But, for myself, I invariably place them with the under side toward the boiler. I never have had any trouble. I claim that a globe valve has no business on a feed-water line anyway; I prefer a good gate valve which will give a straightway opening from the pump to the boiler.

While I do not like globe valves on a blowoff line, I have never had any trouble with them excepting that due to rapid wear of the seat and disk. They are more easily clogged with scale than the gate valve or the regular blowoff cock.

In conclusion, I wish to say that I think with proper care and usage, a globe valve designed for the pressure to be taken care of, will cause very little trouble when installed with the pressure under the seat.

A. A. BLANCHARD.

Oak Harbor, O.

Colors for Station Piping

It was up to me recently to scheme up a color system for a power plant. After looking over the various articles in POWER, I made up the following, which I think has some "system" to it.

All high-pressure or supply lines are painted a solid color; that is, the fittings are the same as the pipe.

The various colors and their meanings are shown by the following table:

COLOR SCHEME FOR STATION PIPING.

Contents of Pipe.	General Color.	Color of Fittings. Supply or High Pressure.	Exhaust. Low-Pressure or Return.	Drips, Drain, Etc.
Steam	white	white	red	brown
Water	slate	slate	red	brown
Oil	dark blue	dark blue	red	
Air	light blue	light blue	red	brown
Fire service	vermilion	vermilion		
Hydraulic pressure	black	black	red	
Ammonia	yellow	yellow	red	brown
Brine	green	green	red	brown

As a further aid in distinguishing what the pipes contain, I had stencils cut with arrows and numbers to indicate the normal pressure and the direction of the flow. The arrows were put on every pipe where the direction of flow was not obvious from the position or location of the pipe. The pressure figures were neces-

sary because of the fact that three different pressures of steam and two of compressed air were carried.

LOUIS TONG.

New York City.

Leaky Floats

In the August 16 issue, E. Granfield asks for information regarding leaky floats.

To my knowledge, there is no successful method of repairing a metal float other than by soldering it.

I suppose the float Mr. Granfield has reference to is a round one. If the float is made of copper, take some muriatic acid, 5 cent's worth will be sufficient, in a drinking glass. Fill the glass one-quarter full and put in some zinc, in pieces small enough to be covered by the acid.

The liquid will boil as soon as the zinc is put in. When the foam rises to the top of the glass, ignite it with a match and allow it to burn off. Do this in the evening and let the glass stand uncovered over night. In the morning the liquid will be as clear as crystal. The solder should be composed of two parts lead and one part good tin.

When ready to solder, have a small brush handy, light some charcoal in a soldering pot and put the soldering iron on to heat. Punch a hole in the float to let the water out. In 99 out of a 100 cases the float leaks at the seam. As it is hard to tell just where the leak is, it is best to melt the ball apart. While the solder is in a melting condition, wipe the joint clean with a rag. Scrape and tin all black spots. Remember, both edges of the joint must be perfectly tinned to insure a tight job. If everything is tinned nicely and no lumps of solder remain on either of the edges, put the halves together again. In putting them together take a strip of tin or copper and force it into the upper part of the seam and pull it around, forcing one part into the other. This done, heat the iron well and with a fine file rub two sides of the point until they are bright; dip the point into the acid for two seconds and then apply it to the solder which will melt immediately and tin the iron. Swab the float all around the seam. The soldering may then be commenced. Do not put too much solder on. Go all around the seam and solder smoothly. Put a small drop of solder here and there on the seam so as to prevent the float from coming apart. When the iron does not melt the solder any more, heat it again, do not try to smear the joint tight, the solder must run freely; otherwise a leak will surely develop in a short time.

After the joint is soldered smooth all around and everything appears to be tight, go over the joint again, this will make it more solid. Solder the socket on and then close up the hole which was punched to let the water out.

Before putting the float back in the tank try it. Push the float into a bucket of water so that the ball is wholly immersed for a while. If time permits, tie the float down over night. If there is no water in the float in the morning, it is safe to assume that the float is perfectly tight.

Remember, a hole only as small as the end of a pin will cause trouble. If all the rules are followed, however, the ball will be tight and as good as or better than a new one. I have repaired several floats. The last one that I repaired has been in use for the last three years. An experienced hand can do the job in about two hours.

WILLIAM L. KEIL.

Philadelphia, Penn.

Chimney Design and Wind Pressure

In the August 16 issue appears an article on chimney design by H. R. Austin. Perhaps the following information may be of service to him and others who read POWER. I believe that Mr. Austin is correct in his statements.

The relation between the mean pressure of the wind against an exposed surface and the corresponding wind velocity has never been worked out with entire satisfaction. Nevertheless, better results can be obtained by using such facts as are known than by attempting to measure the wind pressure directly.

TABLE OF WIND PRESSURES.

Indicated Velocity in Miles per Hour.	Tenths									
	0	1	2	3	4	5	6	7	8	9
	Wind pressure in pounds per square foot.									
0						0.104	0.144	0.190	0.243	0.303
10	0.369	0.433	0.511	0.586	0.666	0.762	0.853	0.949	1.05	1.16
20	1.27	1.38	1.50	1.63	1.76	1.90	2.04	2.19	2.34	2.48
30	2.64	2.81	2.98	3.14	3.32	3.50	3.67	3.87	4.04	4.24
40	4.44	4.64	4.84	5.07	5.27	5.51	5.72	5.93	6.18	6.40
50	6.66	6.89	7.12	7.40	7.64	7.88	8.14	8.43	8.69	8.95
60	9.22	9.49	9.76	10.1	10.4	10.6	10.9	11.2	11.6	11.9
70	12.2	12.5	12.8	13.1	13.5	13.8	14.1	14.4	14.8	15.1
80	15.5	15.8	16.2	16.5	16.9	17.3	17.6	18.0	18.4	18.8
0	19.2									

Experiments were made by the Weather Bureau at Washington to determine the relation between wind velocity and pressure for certain limited conditions by exposing squarely against the wind plates having from 4 to 9 square feet of surface. The deflection produced by the pressure in springs of known strength was continuously recorded by a pencil upon a revolving cylinder of paper. At the same time and on the same sheet of paper was electrically recorded the velocity of the wind as indicated by an anemometer exposed near the pressure plate. The accompanying table gives the results of these experiments.

H. WILKINSON.

Hartford, Conn.

The Selection of an Engine

The articles that have appeared recently under this heading have been very interesting, and it will be still more interesting if the discussion results in some definite information on the subject.

Apparently, Mr. Myers' proof that high piston speeds mean greater economy, misled his critic, and it should be taken that more steam is passed through the cylinder by increasing the number of strokes, the cutoff in the cylinder being kept constant. It would, perhaps, have been more conclusive, and a greater help, had authorities been quoted for the statement that, with two engines of the same class and design, working with the same mean effective pressure and under the same conditions, the low-speed engine would be more economical than the high-speed one. That this is correct is well known, and, no doubt, authorities could be given for it. I would be glad to know of other authorities besides Mr. Willans who have investigated this. Mr. Myers' experience is directly opposed to this, and it would be a great help toward a better knowledge of this subject if he would kindly give the exact data upon which his statements are based. There are very few engineers who are in a position to establish the truth of many of the present-day assertions, and this probably accounts for the fact that, after more than a hundred years' study of the steam engine, there are still some mysteries about this ancient prime mover.

As far as I can ascertain, Mr. Myers is the first to state that the cylinder condensation is directly proportional to the speed of rotation of the engine. Up to the present in this country it has only been suggested that the speed of rotation might affect the cylinder condensation, and there is definite information that a moderate alteration in the speed does not affect the economy of an engine.

I am able to give the following example from actual practice to support this statement, and it is possible to obtain such an example from more than one of the leading engine builders in this country.

In answer to an inquiry for an engine to work under certain conditions, a 15

and 25½ by 30-inch engine to run at 165 revolutions per minute was offered at a guaranteed steam consumption of 11.7 pounds per brake horsepower under the specified conditions. This quotation was followed up by a representative calling upon the firm inquiring for the engine. He was informed that unless the steam consumption were improved his engine would be out of the running. As the result of this information the quotation was amended by offering a 17½ and 30 by 36-inch engine to run at 100 revolutions per minute and a guaranteed steam consumption of 10.8 pounds per brake horsepower under the same conditions as before. Now, these two engines were of the same class and of exactly similar design in every respect; the 30-inch stroke engine was designed for a maximum piston speed of 825 feet per minute and the 36-inch stroke engine was designed for a maximum piston speed of 840 feet per minute. Although the latter engine was quoted at nearly 30 per cent. below the speed for which it was designed, yet the steam consumption was quoted the same per brake horsepower as it would be if the engine were run at its maximum speed under the same conditions and with the same mean effective pressure.

It is interesting to record that the amended quotation was successful in spite of the additional cost of the engine, and this tends to show that the question of economy versus capital cost is considered.

This example has no uncertain quantities in it, and the engine builders guaranteed the steam consumption under the heavy penalty of £500 per pound of steam in excess of this guarantee. It shows that it is not correct to assume that the cylinder condensation is in proportion to the speed, and it also shows that assumptions of the probable effects of certain parts of the steam-swept surface of the cylinder are valueless.

To make this discussion of any practical value, concrete facts only are required, and it would be a great advantage to readers of POWER if, instead of mere statements of experience, the actual data upon which the statements are based be given. I would point out that Mr. Myers is mistaken in claiming economy for the high-speed engine on account of its reduced clearance, for it is just on this point that the low-speed engine scores.

It would be of great assistance to engine designers if some definite information were available of the exact effect, in pounds of steam per brake horsepower, of additional clearance.

The superior economy of the low-speed engine is now most conclusively proved by our workshops, for, now that there is a due regard paid to the cost of coal, the high-speed engine maker is obliged to make the exhaust-steam turbine as well as the standard engine, and the low-speed engine maker is reaping the benefit of being able to supply an engine which easily competes with the overall economy of the high-speed engine and exhaust-turbine combination.

That the high-speed engine has had a great run in this country is not disputed, and in a way it was a natural consequence of cheap coal and a large export trade; the price of coal, however, is now receiving its proper consideration all over the world and the result is that engine builders are now selling both at home and abroad heavier, and more costly, but more economical engines.

JAMES CANNELL.

Stanford le Hope, Eng.

Trouble with Barometric Condenser

I was interested in the description of the trouble that Mr. Elston is having with his Bulkley condenser. A few years ago I worked in a plant which used Bulkley condensers and the same kind of trouble was experienced. The chief engineer was under the impression that it was caused by the tail pipes of all of the engines discharging into a common pipe which discharged into a hotwell, only one hotwell being used. From the sketch shown it is evident that Mr. Elston is having the same trouble with the condenser discharging into an independent hotwell.

I am of the opinion that the trouble is caused by producing a vacuum too rapidly; the action being as follows:

Before the vacuum is produced, the water in the tail pipe is at the same level as the water in the hotwell. When a vacuum is made very quickly, the water in the hotwell rushes up into the tail pipe and the kinetic energy is sufficient to carry it beyond the static barometric level and discharge it into the exhaust pipe. This trouble can probably be eliminated by cutting off the bottom of the tail pipe a little so that only enough water can rush into the tail pipe to fill it about two-thirds full, the remainder being obtained from the supply pipe.

CHARLES R. SUGG.

Wilmington, N. C.

Safety of Boilers

The editorial in the August 16 issue entitled, "Safety of Boilers," contains this statement:

"Overworking a boiler as regards the quantity of steam generated is just as dangerous as overpressure and possible more so, because of the insidious manner in which defects rendering a boiler unsafe are produced by this means."

Considering the number of disastrous explosions which have occurred lately, this warning to users of multitubular boilers should not be forgotten. I have been able to prove the assertion to be correct and probably an account of my observations may be interesting to other readers.

I am chief engineer of an ice factory. We have a 20- and a 60-ton ice machine. The boiler plant originally consisted of two 7x30-foot Lancashire boilers; two 5x15-foot multitubular boilers, and one 5x18-foot Cornish boiler, a miscellaneous collection, but that is another story. We used to run the 60-ton machine at 40 revolutions per minute, making 42 tons of ice per 24 hours with one Lancashire boiler and one multitubular, changing after three weeks' run to the other Lancashire and multitubular. These ran the machine without trouble due to want of steam. But when we worked a Lancashire and the Cornish together, we had trouble in keeping steam.

The earthquake of 1907 threw down the brick chimney, wrecking the boiler roof over the Lancashire boilers and the steam piping. As the boilers were old, I advised remodeling the plant and installing four 6x18-foot multitubular boilers which would occupy very little more floor space than the two Lancashire boilers although they would give much more power.

While the boilers were being imported I put up temporary steam piping to the two 5x15-foot multitubulars and the 5x18-foot Cornish boiler and ran any two at a time, making 30 tons of ice. When I ran the two multitubulars together, I got all the steam required without forcing but when one was shut off for cleaning and the Cornish boiler connected, I had to force the multitubular. (A Cornish boiler cannot be forced with natural draft because the furnace is too small). After doing this for about four months, I noticed that the girth seam over the furnace was leaking. I had it calked and tested. It did not leak when steam was up and the boiler working but only after it had been cut out and allowed several days to cool. This leaking seam bothered me for I could not find the cause. These multitubular boilers had been run by me for the previous ten years and were in good condition.

All this while I was pulling out the old Lancashire boilers and preparing a place for the new 6x18-foot multitubulars. Two came and were erected but, before they could be worked, I had to pull down one of the 5x15-foot multitubulars. We then had the two new boilers, one 5x15-foot multitubular and the Cornish boiler. I started to make 42 tons of ice as before the earthquake, using one of the new boilers and the old multitubular boiler for three weeks, then changing to the other new boiler and the Cornish. In about three months the new boiler which worked in connection with the Cornish began to leak in the girth seam over the furnace. I had it calked and tested. After running for three weeks more it leaked again on cooling; while running it

did not leak. This was a condition that could not be allowed to continue, for calking was of no permanent value. The cause must be discovered and removed. The boilers were new and why should one be troubled in this manner and not the other? The answer came, you are forcing the boiler too much. To prove it, cut out the Cornish boiler, pull it out to make room for the other two 6x18-foot multitubulars and run the three boilers any two at a time. I did so and the leak stopped and has stayed so for the last eighteen months.

From this experience I have drawn the conclusion that the multitubular boiler is an element of danger when forced to generate steam beyond a certain capacity. Supplying more steam than can safely be generated is the short cut to a patched girth seam. When a boiler is forced, this seam expands so much that in contracting the rivets do not contract in the same proportion.

E. Browstorph.

Kingston, Jamaica, B. W. I.

An Emergency Decision

I read with interest the letter by Mr. Bloss under the above caption in the August 23 number. The question may be asked, how long can a boiler be run with any degree of safety when the water is out of sight in the glass? With a vertical water-tube boiler the conditions are more favorable than with other types, as there is less liability of burning the tubes. The tops of the tubes where they expand into the steam drum or header are vital spots in water-tube boilers even if they are reinforced. This is especially true if the bottoms of the drums are exposed to the direct action of the fire. The fusible plug would probably melt with 6 inches of water in the drum and prevent the pressure from rising and causing a serious explosion, but the tube ends are apt to be contracted and cause leakage and the tube sheet or header is in danger of becoming weakened and giving way later under working steam pressure.

Mr. Bloss evidently depended on the fusible plug for safety. He states that everything went well for several days. It seems to me that in this length of time a feed pump could have been connected up temporarily to the boiler. If a pump was not on the premises at the time, it would appear that Mr. Bloss had overreached his mark in trying to make good by endangering life and property in leaving a fireman in the plant at night with improper equipment to operate the boiler. I believe that he should not have started the boiler until the pump was received.

I left a watchman who had a fireman's license to fire some horizontal water-tube boilers one night. On entering the plant the next morning I found him very nervous and the work rather neglected. "Those boilers don't hold much water, do they?" he remarked. During four hours of the night the fires were covered and the damper closed as enough steam could be furnished with the fires in this condition. He had taken a nap during this period. When he awoke the water was out of the glass. The feed water was cold, but could have been heated by allowing the steam to blow through the engine. Instead of doing this he had pumped in the cold water, and as nothing had let go, he concluded that all was well. I opened the back connection door and the hood doors in front and found that a large number of tube caps had been started and were leaking badly. The boiler had to be run for three days and nights in this condition before it could be shut down and the caps reground.

In another case, three nipples connecting the header and the mud drum let go and put the boiler entirely out of service. Had the fire in this case been bright and working, these tubes would have been badly burned.

In still another case, both the fireman and the night engineer went to sleep. When the fireman awoke he found that he could not keep water in one of the boilers, which were of the horizontal tubular type. Investigation showed the fusible plug had melted and the top tubes were leaking badly. A new row of tubes had to be put in at a cost of $300, as the burned ones could not be rolled tight. The boiler was made perfectly tight but the tube sheet was weakened by being overheated.

R. A. Cultra.

Boston, Mass.

In the August 23 issue there appeared a letter by Mr. Bloss on an emergency decision. In my opinion the decision was foolhardy.

There might possibly be occasion for such proceedings, providing the engineer had been around the boiler immediately before the water passed out of sight, or knew the time that had elapsed since the alarm began to blow; if he knew that the columns and gages were working properly, and knew the actual distance from the lower gage cock or glass to the danger line, besides having some idea as to how fast the water would be lowered due to the load. I might add that he should be sure the blowoff valve was all right. Notwithstanding all this, it would not be good engineering but might be tolerated in extreme cases. But as Mr. Bloss says, he was about a mile from the plant when he heard the alarm, and he had no assurance of how long it had been blowing.

Going that distance, getting the fires started, and raising steam about 30 pounds, required considerable time and he had no guarantee that the injector would work immediately after obtaining the required pressure. I think the majority of engineers will agree with me when I say that it was foolhardiness, and such practice should be severely condemned.

Joseph Stewart.

Hamilton, O.

Operating Engineer versus Consulting Engineer

I notice that in his reply to Warren H. Miller, Thomas B. Whitted, in the August 30 issue, says that power plants designed by those whose sole experience lies along operating lines are likely to be of rather inefficient type. This may be true in most cases but, on the other hand, power plants designed by those whose only training is a college education are likely to be conglomerations of hair-splitting theoretical monstrosities entirely unfit for practical results.

What the power-plant owner is looking for is results; what the practical operating engineer is looking for is results. Besides, Mr. Whitted must remember that there are very few operating engineers today whose education lies solely along operating lines. It makes but little difference where a man gets a thing as long as he has it. The time has passed when the operating engineer is content with knowing how to shovel coal and squirt oil.

Of course, consulting engineers are necessary adjuncts in the engineering world; so also are operating engineers. We must not jump at the conclusion that an operating engineer is incapable of understanding the theoretical part of steam engineering. The sooner this idea is laid aside and the consulting engineer comes to the conclusion that he is not the whole thing and the operating engineer conceives the fact that he can learn some things worth knowing from the consulting engineer and the consulting engineer realizes the fact that his dignity does not suffer by considering a suggestion offered by the operating engineer, the sooner will the plant owner secure better results.

To show the inconsistency of some consulting engineers I will relate an incident which came to my notice recently. When a certain consulting engineer was figuring on a plant I asked his opinion regarding centrifugal pumps for boiler-feed purposes. He remarked that they were only a fad. But I noticed that when the specifications came out centrifugal boiler-feed pumps were specified. From this it would seem that this particular consulting engineer figures on fads for his clients.

I also asked him about his opinion of boiler-feed regulators. He said that he would not recommend such things as he would not have a man in a boiler room

who could not look after the water properly without such things. This man did not consider the economy of such appliances at all but condemned them merely because they took some of the work off the boiler-room attendant. I know from practical experience that good boiler-feed regulators will pay for themselves several times in one year.

The sooner steam-plant owners learn that their regular engineer has as great an interest in their plant as anyone and is best qualified to pass on contemplated improvements, the better off they will be. While he may not be qualified to design a plant, he is best qualified to judge whether the apparatus designed by the consulting engineer is the proper thing for his plant or not.

When the consulting engineer and the steam-plant owner acquire sense enough not to feel that their authority has been invaded or their dignity lowered by calling in the operating engineer, results which will be more nearly ideal will surely be secured.

H. R. ROCKWELL.

Mt. Vernon, Ill.

Rope Transmission Problem

Replying to Mr. Kirlin's question in the August 30 issue regarding the "Rope Transmission Problem," I believe that one sheave being ¼ inch smaller in diameter than the other will impair the efficiency. Assume that the two sheaves are not keyed to the shaft but can revolve around it, each sheave independent of the other.

Let A denote the sheave whose diameter is 9 feet, and let B denote the sheave whose diameter is 8 feet 11¾ inches.

The revolutions of the driven sheave equals the revolutions of the driver multiplied by the diameter of the driver and divided by the diameter of the driven sheave.

Assume that the driving sheave, which is 14 feet in diameter, makes 1000 revolutions per minute; then, the revolutions of

$$A = 1000 \times \frac{14}{9} = 1555.5$$

The distance that the rope will travel per minute is

$$1555.5 \times 3.1416 \times 9 = 43{,}980.46 \text{ feet.}$$

For the same number of revolutions of B the rope will travel

$$\frac{1555.5 \times 107.75 \times 3.1416}{12} = 43{,}878.06 \text{ feet.}$$

Under the conditions cited by Mr. Kirlin, for every 1000 revolutions of the engine pulley there is a rope slippage of 102.4 feet on either A or B. Now, proceed to determine on which sheave the slippage occurs. The horsepower transmitted to the sheave is

$$\text{Horsepower} = \frac{2\pi D \times RPM\,(T - F)}{3 \times 550}$$

where,

T = Tension on driving side of the rope, in pounds;

F = Centrifugal force of the rope in pounds;

D = Diameter of the sheave.

From this formula it will be noted that the larger D is, the greater will be the horsepower transmitted. Therefore, it follows that the rope will slip on the sheave which has the least horsepower transmitted to it, which in this case is B, the smaller sheave. This means that the ropes which travel over this sheave will wear faster than the others.

G. F. ATWATER.

South Norwalk, Conn.

Soot and Ashes on the Tubes

Referring to William Westerfield's article in the August 23 issue of POWER, I take exception to some of the impressions which it left with me.

In the plant where I am employed there are three 250-horsepower horizontal water-tube boilers, two of which are in one setting, the other one being in a setting by itself. The distance between the double and the single setting is about 36 inches. Oil is used for fuel. The oil has a specific gravity of 14 degrees Baumé and contains an average of 8 per cent. basic sediment and moisture. The plant is operated continuously. Each boiler is cut out and cleaned once every 30 days. This is done with a mechanical cleaner. The cleaner is run through the tubes from the front end, the space required for this purpose being about 8 feet. The soot is blown from the tubes by steam pressure applied through the front sheet, at the side of the setting and through the rear. It is quite true, however, that this alone will not keep the tubes clean and that some other work must be done on them from time to time.

As to Mr. Westerfield's "assumed" and "actual formation of soot on the tubes," he claims that Fig. 1 in his article is an extreme case to be found only "in some Arkansas sawmills." While our plant here can hardly be classed as an "Arkansas sawmill," I am forced to confess that the boiler tubes are in that condition and we do not consider it as an extreme either. In fact, close observations in practical experience lead me to class Mr. Westerfield's Fig. 1 as representing a more common condition than his Fig. 3, which he designates as representing actual conditions.

If there were nothing to contend with but the soot, Mr. Westerfield would be right in speaking of Fig. 1 as representing an assumed formation, but so long as ash is one of the products of combustion it also will have to be considered. The ash which concerns us is that which is in the form of dust. The construction of the horizontal water-tube boiler is such that the hot gases are baffled by the tubes being set staggered. This gives a condition which produces a dust trap on the top of each tube where the gas is not in rapid motion.

B. F. HARTLEY.

Tipton, Cal.

Engineers' Wages

In the September 13 number I find a very interesting piece of literature on "Engineers' Wages," the author of which is certainly not bashful about letting us know what rot an engineer (?) can dish out when writing about something beyond his accustomed horizon.

His daring to distinguish the engineer from the "stopper and starter" is equaled only by his innocent childishness when telling about the good, satisfied engineers working for $45 per month. I can imagine his wrath at finding a lot of progressive and aggressive "stopper and starters," who, by the way, probably would not submit to his teachings, working for $85 per month, and kicking at that.

His remarks about "unions" show an utter ignorance of the fundamental principles of self-preservation as well as a good-size affliction of what is generally termed "swelled head." Intelligent people are not opposed to unions. Our forefathers fought for the Union; they knew by dear experience that divided they would fall. The lawyers have their bar associations to uphold their standing and, incidentally, prices. The medical associations are formed for the same purposes, also the civil and mechanical engineering societies and similar organizations. Carpenters, masons, machinists, etc., bettered their financial and social conditions through unionism. Why should the operating engineer be the only "fool," to try to better his social standing single handed? The National Association of Stationary Engineers is a living proof of the fallacy of standing on your own merits, and allowing your employer, some say exploiter, to be the jury of one to fix the dollar value of these merits.

At present, there are a lot of good engineers (not the $45 kind) hard at work to form and organize an uptodate association, "The Institute of Operating Engineers," which, as far as I can help, will be a union for education, elevation and protection.

I am proud to be a charter member of this really progressive institution and hope that all readers of POWER will step nearer and investigate, and help to build up a powerful organization for *our own* benefit, an organization that will enable us to claim and secure by superior education and stick-together-activity the professional and social recognition the uptodate stationary engineer has deserved for so long.

H. F. HEYRODT.

Bridgeport, Conn.

Inquiries of General Interest

Questions are not answered unless accompanied by the name and address of the inquirer. This page is for you when stuck—use it.

Selection of a Steam Boiler

What are the three prime requisites to be considered in the selection of a steam boiler?

S. R. B.

There are many things to be considered in the selection of a steam boiler, among the most important of which are safety, durability and the economy of operation.

Vacuum

What is meant by vacuum?

M. V. E.

By definition, vacuum is empty space. In steam-engine practice the term refers to space in which the steam is or has been condensed to a greater or less degree, and is never really vacuum.

Pressure in Steam Boilers

Is the pressure equal in all parts of a steam boiler?

P. S. B.

No, the pressure on the bottom of the boiler is greater than that on the top by the weight of the water. The pressure of the steam is exerted equally in all directions on the water as well as on the shell, but the weight of the water is supported by the bottom of the boiler only, and in addition to the pressure of the steam.

Foaming Boilers

What is foaming and what will prevent it?

F. I. B.

Foaming in steam boilers is a violent frothy ebullition which carries water in the form of foam to the steam space and the main steam pipe. It can be prevented by using clean water in a clean boiler.

Object and Size of Safety Valves

What is the object of a safety valve and how is the right size for any boiler found?

O. S. S.

The object of a safety valve is to prevent the pressure in a boiler from rising higher than some predetermined point, by opening at this pressure and allowing the excess of steam to escape to the atmosphere. Its capacity to relieve the boiler is calculated from the amount of water that may be evaporated by the boiler. It is customary to allow ⅓ square inch of safety-valve area for each square foot of grate area when the valve is "spring loaded" and ½ square inch of valve area per square foot of grate when the valve is of the ball and lever type.

Air and Moisture

I wish to pass a current of air over a steam coil for the purpose of drying it. How hot will the air have to be to be thoroughly dried?

A. A. M.

Heating the air will not dry it at all, but, on the contrary, will increase its capacity for absorbing moisture. Air can be thoroughly dried by freezing.

Comparative Evaporation

How may the capacity of a boiler taking feed water at 70 degrees temperature be compared with one which is being fed with water at 180 degrees?

B. E. C.

By reducing both to the standard of equivalent evaporation. Subtract the number of heat units in a pound of feed water at the given temperature from the number of heat units in a pound of steam at the temperature at which it is made and divide the difference by 970.4. The quotient multiplied by the number of pounds of water actually evaporated will give the equivalent evaporation from and at 212 degrees, that is, the weight of water which would have been evaporated from feed water at 212 degrees into steam at atmospheric pressure. By doing this with both boilers their respective work may be compared.

Multiported Valve Lead

In setting the Allen valve, should it be given the same lead as the plain slide valve?

M. V. L.

Yes, but owing to the fact that the Allen valve is double ported it should have but one-half the apparent lead or port opening as a plain slide valve, as the lead is really double what it appears to be, because the steam is admitted to the cylinder past two valve edges instead of one.

Compound Engine Cylinder Ratio

In a noncondensing compound engine the high-pressure cylinder is 18 inches in diameter. What should be the diameter of the low-pressure cylinder, if 150 pounds steam pressure is used?

E. C. R.

For an equal distribution of work between the cylinders the number of expansions in each should be equal. Expanding to five pounds pressure the number of expansions will be 8.25. The cylinder ratio should be the square root of the number of expansions

$$\sqrt{8.25} = 2.87$$

Therefore the area of the low-pressure cylinder should be 2.87 times that of the high

$$2.87 \times 254.47 = 730.32$$

square inches.

To find the diameter of a circle from its area, use the formula

$$D = 2\sqrt{\frac{a}{\pi}}$$

Substituting, the formula reads

$$\text{Diameter} = 2\sqrt{\frac{730.32}{3.14}} = 30.5$$

inches, the proper diameter for the low-pressure cylinder.

Does the Crosshead Stop?

Does the crosshead of an engine stop at the end of the stroke, and, if so, how can it be proved?

C. H. S.

During one stroke the crosshead moves in one direction and during the next it moves in the opposite direction. If it does not stop there will be an interval between the changes in the direction of its movement in which it must travel in two directions at the same time, which is absurd.

Temperature of Gage Spring

Why is the pipe leading from the steam gage to the boiler always cold even when there is 100 pounds pressure in the boiler?

T. G. S.

The pipe leading to the gage is always put up with a bend in it which serves as a pocket in which water collects from condensation the first time the boiler is fired. It always stays in the pipe and prevents live steam from coming in contact with the gage spring and causing it to register incorrectly as it would if exposed to the temperature of the steam in the boiler.

Running Engine Single Acting

Can I run a Corliss engine with one exhaust valve removed and the steam valve blocked shut while a new stem is being fitted to the exhaust valve?

J. I. D.

Yes, if the flywheel is heavy enough to carry the engine through one-half revolution and over the center with full load.

Issued Weekly by the
Hill Publishing Company
JOHN A. HILL, Pres. and Treas. ROB'T MCKEAN, Sec'y.
505 Pearl Street, New York.
150 Michigan Avenue, Chicago.
6 Bouverie Street, London, E. C.
Unter den Linden 71—Berlin, N. W. 7.

Correspondence suitable for the columns of POWER solicited and paid for. Name and address of correspondents must be given—not necessarily for publication.

Subscription price $2 per year, in advance, to any post office in the United States or the possessions of the United States and Mexico. $3 to Canada. $4 to any other foreign country.

Pay no money to solicitors or agents unless they can show letters of authorization from this office.

Subscribers in Great Britain, Europe and the British Colonies in the Eastern Hemisphere may send their subscriptions to the London Office. Price 16 Shillings.

Entered as second class matter, April 2, 1908, at the post office at New York, N. Y., under the Act of Congress of March 3, 1879.

Cable address, "POWPUB," N. Y.
Business Telegraph Code.

CIRCULATION STATEMENT

During 1909 *we printed and circulated* 1,875,000 *copies of* POWER.

Our circulation for September, 1910, *was* 134,000.

October 4..... 33,000

None sent free regularly, no returns from news companies, no back numbers. Figures are live, net circulation.

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Power and The Engineer

Editorial

The Cost of Power

It has been so frequently repeated that an engineer should know more about the art and science of steam engineering than any other one subject, that another repetition of the statement seems almost, if not quite, out of place. But the rate at which the central station is placing contracts for furnishing current where it is evident that the local plant has a manifest advantage when all of the factors are considered, shows that the engineer is handicapped by his inability to show the proprietor that central-station service is not the cheapest available.

In New York City the engineer is crowded harder than elsewhere in the country by the competition of the central station. But here too he has the best opportunity to show that he can "make good" if he so desires.

It is not necessary to say that the man in direct charge of an isolated plant should know just what can be accomplished in that plant. It is self-evident. But there are hundreds of plants in the city where the engineer has no more idea of the actual cost of the service he furnishes than he has of the plan of salvation. But the central-station solicitor has. He knows how much coal is burned and the amount of light and power furnished. He knows the payroll and the cost of extra work. He has all the data at his finger tips which he uses with a glibness of tongue and the convincing manner of one who knows, and knows that he knows.

Talking in a logical way about things with which he is familiar, and reasoning directly from the coal bunker and water meter to the switchboard and thence to the proprietor's pocket is not the average engineer's long suit, and in competition with the trained central-station man he appears at a disadvantage. But if he knows what he is doing and can show in plain English and correct figures the coal cost, the labor cost, interest and depreciation charges, water rates and repairs there will be little danger that the central station will ever take over his plant, for he will generally be able to show that he can produce electric energy at a lower price per killowatt-hour than any central station, however well equipped, will furnish it.

If any New York engineer feels that the task of proving what can be done with his plant is beyond the limits of his ability, the general committee of the subordinate associations of the National Association of Stationary Engineers of Manhattan and the Bronx, is at his service.

This committee has a complete line of apparatus suitable for power-plant testing, and is prepared at any and all times to make an exhaustive test of any plant operated by a member of the association.

If at any time it is intimated that heat, light or power may be bought for less than it costs to make it at home, the engineer is, if he does not already know, in a position to have determined for him the exact cost and the product of every department under his supervision. From the results of tests made by the committee tabulated statements can be made which may be submitted for comparison with the proposals of the central-station solicitor. The committee is at the service of any member of the National Association of Stationary Engineers. If an engineer who is not a member of the organization needs the service, the proper course for him to pursue is obvious.

Newspaper Accounts

The following extracts from the account of an explosion in a power house in Cambridge, as described by a Boston paper, illustrate how accidents are often misrepresented by the daily press and confirm the statement that anything will do to fill up the news columns.

The report commenced by saying that a boiler exploded, seriously injuring one man and endangering the lives of six others, also badly wrecking the building. A little further it said that the boiler was the largest of four in the plant. Now, after giving the impression that it was a boiler which exploded, the next few lines contained the statement that "one section of the engine shot upward, tearing a hole in the roof." One of the officials, whose business is to issue such reports, is quoted as saying, "the explosion was confined to a receiving engine having a capacity of 2700 kilowatts." And, by way of further explanation, the account says that "the engine which exploded is used to receive the high-tension steam where it is transformed into low tension. It is thought that the capacity of the engine was overtaxed."

Now, for a large number of people this account is sufficient, but for those who would be interested in knowing the real facts of the case, it is extremely inade-

quate and confusing. There is no doubt that something was overtaxed, but it might as well be the memory or the imagination of the reporter as anything else, for, while the reader is thinking about a boiler explosion, he is suddenly switched over to the consideration of an engine breakdown, and a receiving engine at that, which might be run by electricity as it is measured in kilowatts. One would also be led to believe that it was a new form of transformer for steam which is measured by tension instead of pressure.

What actually happened was this: The receiver between the high- and low-pressure cylinders of a compound engine burst, and caused all the trouble. If there is anything about a receiver that refers to boilers, kilowatts or transformers, it would be interesting to know it. If a newspaper would send one of the firemen in its own steam plant to write up such accidents they would in most cases give a better account than the one described.

A Head for the Steam Plant

Operating a power plant twenty-four hours a day, week in and week out, is much different from operating the same size of plant but ten hours a day. Not only is the wear and tear on the engines, boilers and auxiliaries greater, but the fact that two, and in some cases, three sets of men are employed adds greatly to the difficulty in maintaining smooth operation. Every steam plant requires a head—someone who is responsible for all that takes place, whether he is present or not.

One of the difficulties encountered in power plants is that the men in one watch are likely to shift the responsibility for some mishap to the men of the other watch, if it is possible. Take, for instance, the case of keying up an engine. It is a simple matter, but considerable trouble has been caused by improper keying which has not been reported to the relief engineer.

In one case, an engineer decided that the main pillow block of an engine required taking up and at the end of the run proceeded to carry out the work, but he neglected to notify his relief of what had been done. When the relief engineer started the engine, the bearing began to heat badly, but it was noticed and relieved in time to prevent the damage to the bearing. If the engineer had not been watchful the engine would doubtless have been disabled, and all because the relief engineer had not been notified of the adjustment.

There should be one man to look after such work as keying the engine, and that man should, of course, know how to do it. Furthermore, when work of this nature has been done the force coming on duty should be notified. Even the oiler should be told to watch such-and-such a bearing more closely than usual in order to prevent heating in case the box or bearing has been too finely adjusted.

In moderately small plants running on a twenty-four hour basis trouble is frequently caused by either the day or night engineer adjusting the engine to his liking. Under all circumstances the day engineer is the one to make all adjustments. The night engineer should not touch the engine, unless it is something that is absolutely necessary in order to keep it running.

If the night engineer thinks he knows more about an engine than the day man, he should keep his thoughts in the background, and wait until he gets a day run before he tries to put his knowledge into practice. The day man is really the chief and should be so considered by all employed in the plant.

The work in the boiler room also demands a head. Because a certain routine has been carried on for years does not mean that it will be continued. A change in the position of a fireman may cause trouble galore. In a certain plant the head fireman resigned and the next best man was put in charge. One of the ten boilers was washed out each week and this fireman had assisted in the work for years.

In promoting him the chief engineer assumed that the washing of boilers would go on as formerly and, therefore, gave the matter no personal attention. For some reason the new head fireman did not have a boiler washed for four weeks, so that one boiler was run for fourteen weeks without being washed out. At the end of that time it bagged so badly that the fireman's negligence was discovered by the chief engineer.

These are only a few of the many instances that might be mentioned in showing that the power plant requires a chief who will keep in touch with the routine work and see that everything is properly done.

Technical Education

The feeling of contempt so often evinced by the practical man toward the young technical graduate is largely due to too much being expected of the latter through a misconception of the aims and limitations of the technical school. Within the past ten or fifteen years, engineering methods have undergone a vast change, and their scope and magnitude have increased to such an extent that the technical school, in order to keep pace, has been forced to revise its curriculum. The fact is now appreciated that there exists a close relation between the technical school and the engineering profession, but that in order to succeed, the former must base its work upon a thorough understanding of this relation. This is now being accomplished through a study of industrial conditions, with courses arranged to fit these conditions.

Formerly too much emphasis was placed upon the manual features of the curriculum, and attempts were made to accomplish in the laboratory what could be done only in the factory. The technical schools do not attempt to teach trades, and are now breaking away from the manual-training idea, giving only such shopwork as will enable the student to become familiar with modern methods of production. This seems a wise expedient, as the student could not become an expert mechanic even within the maximum amount of time that was formerly allotted to shopwork, and this excess time was at the expense of other subjects essential to modern engineering. Further, it is questionable whether one man out of ten would have occasion later in life to make use of his handiness with tools, and such men can acquire this skill in shops outside of the school. Many of the large manufacturing and operating companies are recognizing this fact and have established apprenticeship courses to supplement the course in the technical school. With an engineering foundation to build upon, they supply the necessary practical training to make the men of greatest use in their particular field.

Another feature that the engineering schools are breaking away from is specialization. This is largely due to two causes: First, the average student does not know what particular line of engineering (this does not apply to the two great divisions, civil and mechanical engineering) he will follow, and as opportunity is the deciding factor in such cases, he is apt to go into an entirely different branch from that which he studied, in which case much of the time devoted to the details of a specialty is wasted. Second, technical schools cannot give courses in special branches that would be equivalent to the knowledge gained by several years' practical experience in those branches, but they can give a broad training in the fundamental principles of engineering, embodying a familiarity with other branches, which will enable the individual to make the most of his future experience—the training which no amount of experience in a special branch could supply.

It is a fallacy to suppose that a successful technical education depends upon book knowledge. Textbooks are only a means toward an end, and their contents are forgotten soon after the class room is forsaken for the business or construction office. The aim of the technical school is to give a training that will enable the student to systematically attack and solve problems as they arise, to promote a faculty for securing information along any line when needed, and to conceive, organize and direct extensive engineering projects, as well as industrial enterprises.

New Power House Equipment

Watts Feed Water Regulator

This feed water regulating system will operate with pressures varying from 5 to 200 pounds, and depends upon the variation of water level, and the steam pressure in the boiler for its action.

It consists of a copper ball which is secured to an L-shaped casting in which two passages have been provided, one for water and one for steam, as shown in Fig. 1.

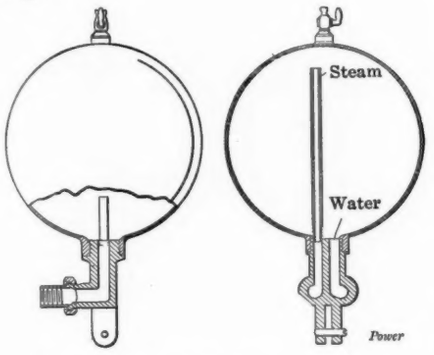


FIG. 1. SECTIONAL VIEWS OF BALL

This copper ball and L connection are attached to the lever of the feed water regulating valve which is placed in the feed pipe line leading to the boiler, as shown in Fig. 2. The ball is also connected to two pipes running to the water column. The pipe leading from the passage connecting with the small vertical pipe in the ball is connected at a point in the water column at which the water level is to be maintained. The water pipe is connected to the water column at a lower point.

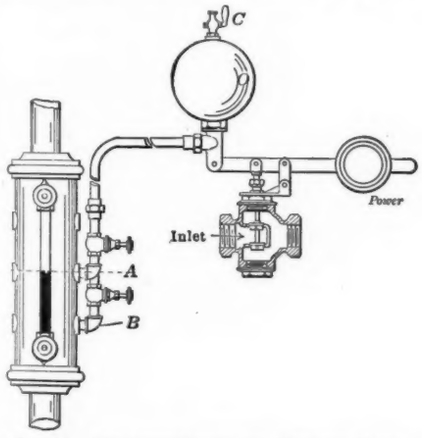


FIG. 2. ARRANGEMENT OF COPPER BALL, VALVE AND PIPING

The arrangement of the ball and lever is shown in Fig. 2. The regulator may be placed at any distance from the boiler by using longer connecting pipes. The copper ball must always be placed above the water line in the boiler.

What the inventor and the manufacturer are doing to save time and money in the engine room and power house. Engine room news.

A sectional view of the balanced regulating valve that controls the feed-water supply is shown in Fig. 3. If conditions are such that the weight and ball lever cannot be attached to the regulating valve, connection can be made by a chain, as shown in Fig. 4.

The general arrangement of the regulator when attached to a boiler is shown in Fig. 5. The pipe *A* is ⅜ inch in diameter and is connected to the water column or to the boiler at the desired point of water level. The pipe *B* is ⅜ inch in diameter and is connected to the water space of the water column or any point below the water line. A ⅜-inch valve is placed in each of the pipes *A* and *B*, near the connection on the water column, so as to cut out the copper ball from the water pressure if necessary. The valve *M* is placed in the feed pipe above the water line of the boiler and the weight *G* is placed on the lever so that it will overbalance the ball *O* when it is empty. The air valve *C* allows air to escape when firing up a cold boiler.

If the water in the boiler or water column is below the pipe *A*, or the water line, the ball *O* will remain empty; also the valve *M* will be opened, because the weight *G* is heavier than the copper ball.

When sufficient water has passed into the boiler to cover the entrance to the pipe *A*, the water will rush into the ball

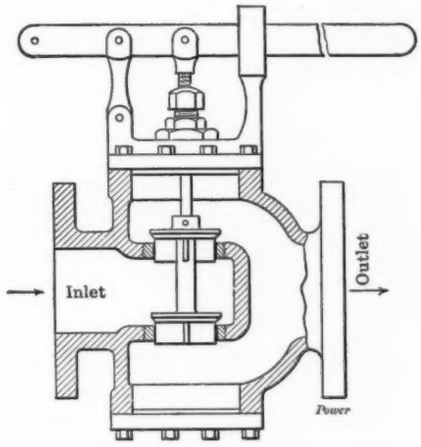


FIG. 3. BALANCED REGULATING VALVE

O, and when a sufficient quantity of water has entered the ball, it overbalances the weight *G* and at the same time closes the valve *M*. The valve M will remain closed until the water in the boiler or water column falls below the pipe *A*, when the water will immediately be discharged into the boiler and steam will take its place. The ball *O* will then be overbalanced by the weight *G* and the ball in rising will open the valve *M* and admit more water to the boiler. The valve *M* will remain open until the boiler is again filled up to the point indicated by *A*, when the operation will be repeated and continued indefinitely. The pipes *A*

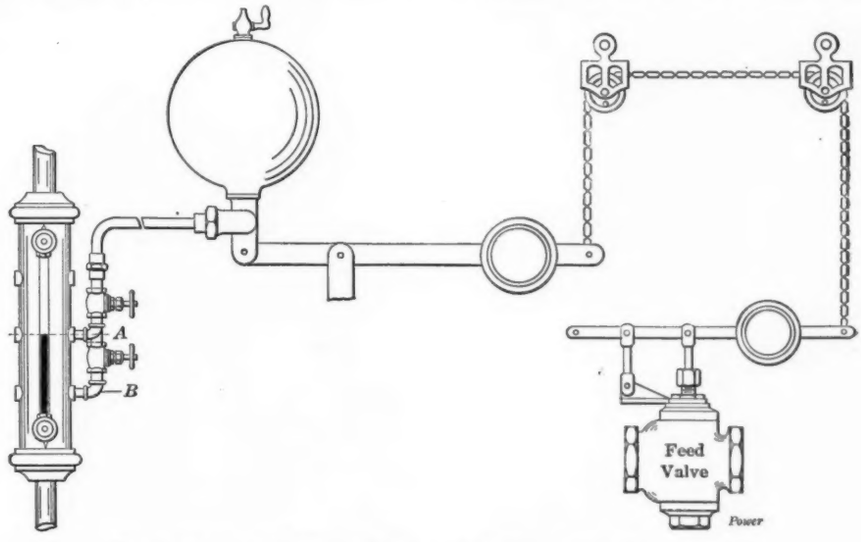


FIG. 4. REGULATOR CONNECTED WITH FEED VALVE BY CHAIN

and *B* are 4 feet long and, being of small size, are flexible.

The water pressure in the feed pipe, which builds up as soon as the valve is closed, is taken care of by a pump regulator valve shown in section at the left of Fig. 5.

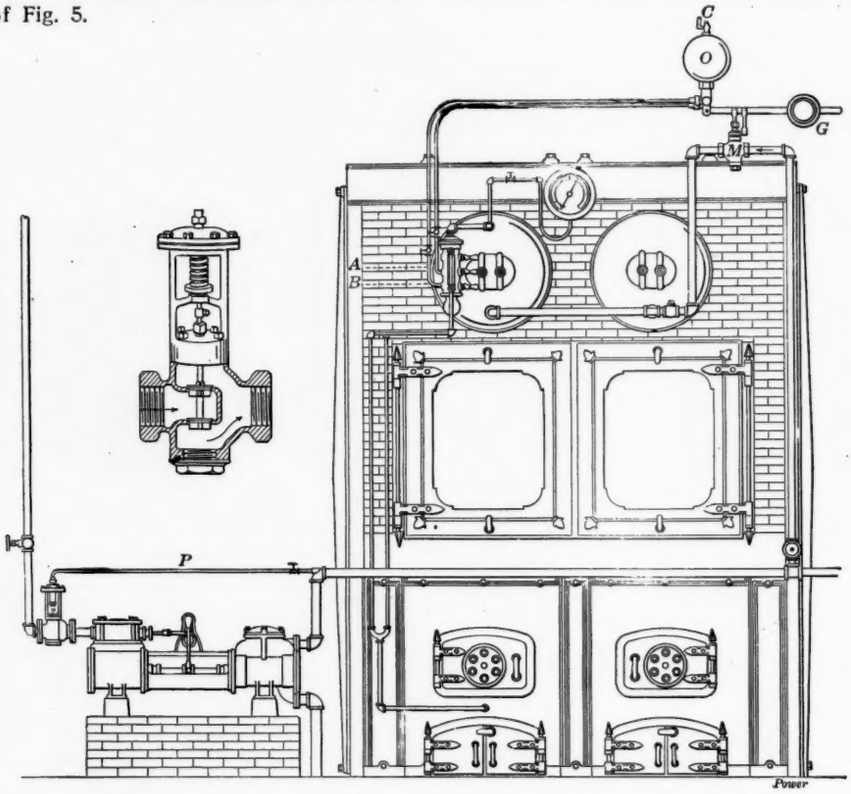


FIG. 5. REGULATOR CONNECTED TO A WATER TUBE BOILER

This valve is balanced and is operated by the water pressure in the pipe, which is exerted against a diaphragm placed above a coiled spring. When the water pressure on the diaphragm overcomes the spring, the balanced valve in the steam pipe line feeding the pump is closed. As soon as the water pressure is reduced, the spring acts and lifts the valve disk from its seat.

The system is automatic in action and requires practically no attention. This regulator is made by the Watts Regulator Company, 250 Lowell street, Lawrence, Mass.

Fugo Grease Remover

This is a liquid preparation made to remove burnt oil or grease from the metallic parts of an engine. It is applied with a small paint brush or a small cotton dauber. After it has been spread over the grease evenly and a few seconds allowed for action to take place, the hardest grease will disappear, it is claimed. After the surface appears bright it must be wiped with a piece of waste.

The fluid acts quickly and ceases to act as soon as the grease is removed, it is said.

This fluid is made by the Fugo Manufacturing Works, Collegeville, Penn.

Osborne High Pressure Valves and Fittings

These specialties are designed to take care of high-pressure work and are suitable for ammonia, steam, either saturated or superheated, air, oil, gas or hydraulic service. The valves are made in both globe and angle varieties and, it is claimed, contain a feature of construction whereby cutting or wire drawing of the seat is practically eliminated.

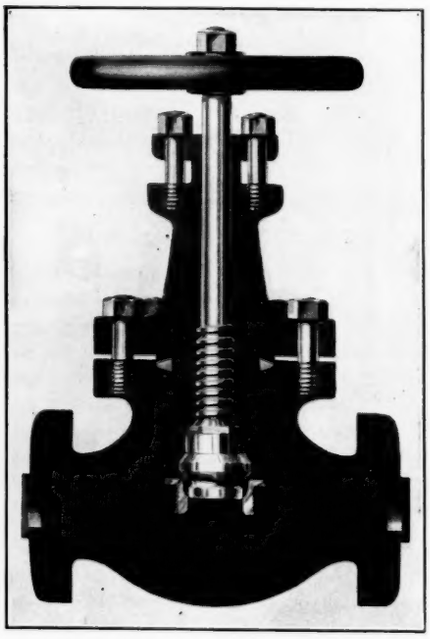


FIG. 1. OSBORNE GLOBE VALVE

Fig. 1 shows a globe valve in section. The disk of the valve carries an extension which closely fits the machined inside surface of the seat. With the valve partly open, the disk extension throttles the flow of steam, transferring the cutting from the seat to the cutting edge provided for this purpose. This prolongs the life of the valve and eliminates wear on the disk, it is claimed.

FIG. 2. RADIATOR VALVE

No considerable flow can take place when either opening or closing the valve, until the disk and seat are far removed from each other, thereby eliminating wire drawing at the place where it would affect the tightness of the valve.

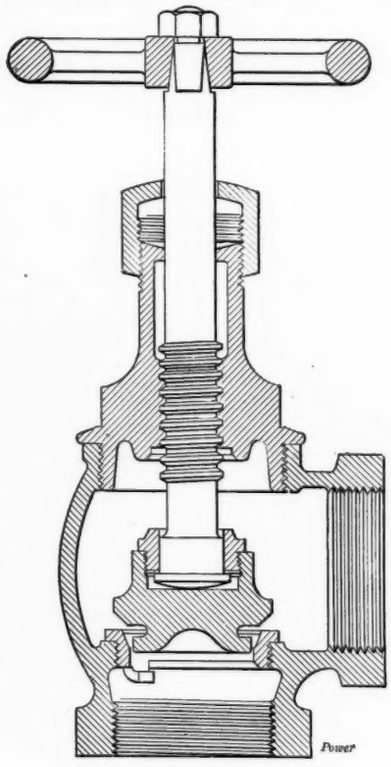


FIG. 3. ANGLE VALVE

Fig. 2 shows an application of the same principle to a radiator valve, the disk extension fitting closely to the inside surface of the valve seat until the valve itself is opened sufficiently so that wire drawing cannot occur.

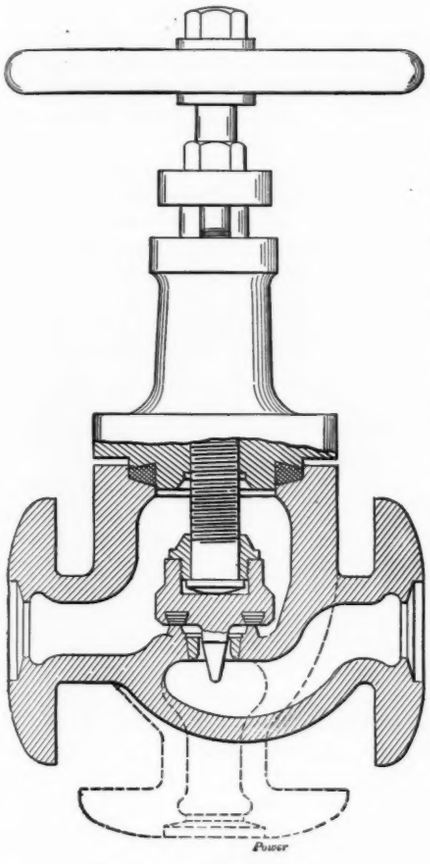


FIG. 4. EXPANSION VALVE

Fig. 3 is a sectional view of an angle valve having the same design of seat and valve disk. A modification of this valve is built for expansion service on refrigerating lines. A section of one of these is shown in Fig. 4. The

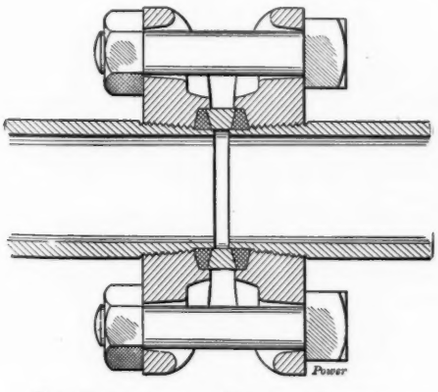


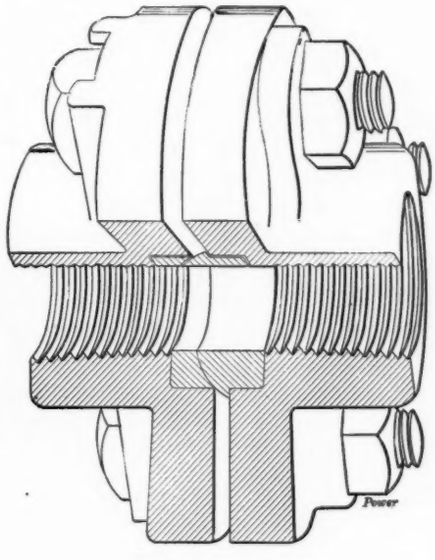
FIG. 5. STANDARD FLANGE COUPLING

extension in this case extends through the expansion ring and when the valve disk is raised the pressure across the seat is balanced, transferring the cutting to the swivel extension and expansion ring. These two parts are hardened steel, thus reducing the cutting to a minimum.

A standard flange coupling is shown in Fig. 5. In making this joint a pipe is threaded a considerable distance from the end and a flange screwed on loosely. The gasket, made of fiber or other composition suitable for the service, fits in a recess in the flange and fits closely around the pipe. A metallic ring is placed between the flanges, covering the junction between the ends of the pipe, making an unbroken metallic surface on the interior. This construction entirely incloses the gaskets, two of which are used on each joint, making it impossible for them to blow out, and enables the joint to be made repeatedly with the same gaskets, it is claimed. As no pressure is placed on the threaded connections, no soldering or sweating is necessary. Further, no welding or facing of the pipe ends is required as they do not touch when the joint is made up. Owing to the construction of the flange it is claimed that absolute alinement is not necessary and that the joint will stand a large amount of vibration and expansion. These valves and fittings are made by the Osborne High Pressure Joint and Valve Company, "The Rookery," Chicago, Ill.

The Dart Flange Union

This flange union consists of two threaded flanges which are screwed onto pipe lengths. One flange is fitted with a female seat, the other with a male seat; both are made of bronze, ground true on the bearing surface. The seats are held together by suitable bolts, as shown in the illustration.



DART FLANGE UNION

This kind of flange coupling permits of connecting pipe lengths when not in line. This flange union is made by the E. M. Dart Manufacturing Company, Providence, R. I.

Pyramid Double Acting Pump

The Goulds Manufacturing Company, Seneca Falls, N. Y., have recently placed

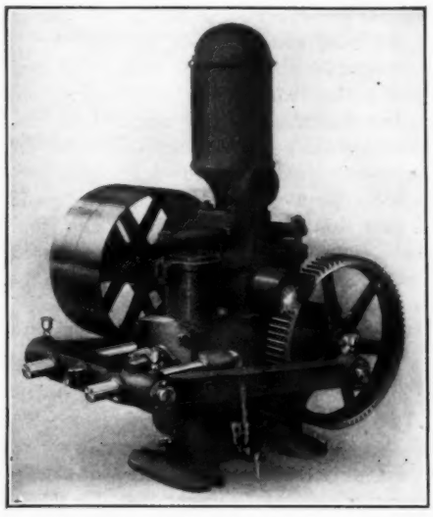


FIG. 1. PYRAMID DOUBLE-ACTING PUMP

on the market the "Pyramid" double-acting piston pump.

The base, cylinder, bearings and one cylinder head are cast integral, thus giving stiff and rigid construction. One cylinder head is cast on the cylinder. As

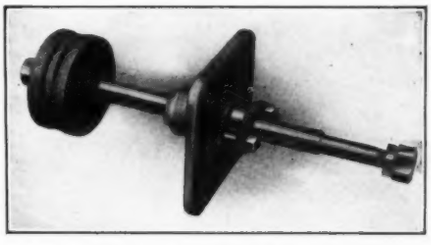


FIG. 2. HEAD AND PISTON

the bearings are cast as a part of the cylinder, the gearing is kept in line.

The cylinder is brass lined, and is provided with large suction valves, placed just below the bore of the cylinder; these

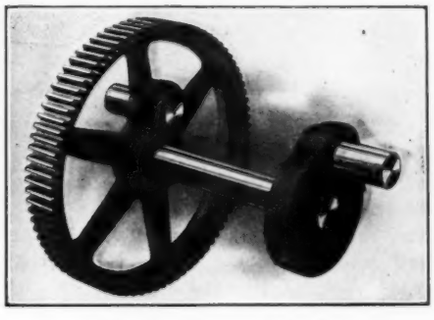


FIG. 3. PINION SHAFT

valves are easily accessible through two handholes, the covers of which can be removed by backing off one bolt.

The discharge valves are placed above the cylinder and are reached through

handholes located on both sides of the air chamber. An exterior view of the pump is shown in Fig. 1.

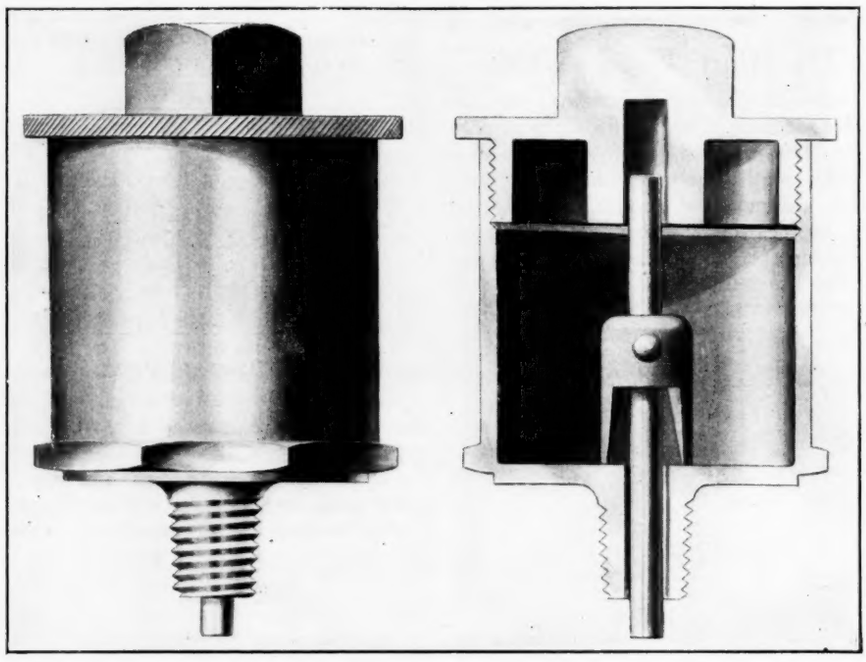
Two steel rods act as a guide for the crosshead. The stuffing box on the cylinder head is provided with a bolted bronze gland. Fig. 2 shows the design of the head and piston.

The power end of the pump consists of a pinion shaft, Fig. 3, supported on a long, double bearing, placed in the center of the pump directly under the air chamber. No outboard bearing is required. Both gear and pinion are made of cast iron, and the teeth are cut from the solid metal. The pump is geared 5 to 1, and is suitable for work up to 75 pounds pressure.

The Jackson Crank Pin Grease Cup

Two views of this grease cup are shown in the accompanying illustration. As shown, by the sectional view, the cup is fitted with a plunger which works up and down, carrying enough grease to lubricate the crank-pin brasses.

This cup can be attached to the strap of any crank rod. The stop shown in the sectional view prevents the plunger



JACKSON CRANK-PIN GREASE CUP

rod from dropping into the oil hole in the crank pin, if such hole exists. The cup requires no regulation and does not feed except when the engine is running. It is made by Richard D. Jackson, 96 Warren street, New York City.

Within 175 miles of Boston, declared an expert before the Massachusetts gas and electric light commission, water capable of yielding 300,000 horsepower is going to waste. One does not have to go into the West to apply conservation.

Meeting of the Central States Water Works Association

The Central States Water Works Association held its fourteenth annual meeting at the Claypool hotel, Indianapolis, Ind., on September 20 to 22. Although the attendance was not large, some interesting papers were read, among which there was one on the subject of rewarding fireman for efficient service, by F. W. McNamee, of Wabash, Ind. In the plant with which he is connected he said that the question of combustion had been investigated and it was found that the results they were getting were bad. Analysis of the flue gases developed the fact that the percentage of carbon dioxide was running from 4 to 5, and sometimes 6 per cent. After studying the problem for some time, a scheme was worked out of paying the fireman on a bonus basis. They are now paid on a flat rate per hour, plus a bonus depending on the percentage of carbon dioxide in the flue gases. Each boiler is supplied with a five-gallon glass bottle, the top being connected through brass pipe to the breeching, just under the stack damper. This bottle has in the bottom of it a brass valve containing a calibrated plug so that in twelve hours it will just empty itself, the water dripping out. The bottle is filled with water at the beginning of a run and on top of the water is placed a small amount of kerosene to prevent solution of carbon dioxide by the water. Each fireman, when he goes on duty, puts on as many bottles as there are furnaces for him to run and starts them to dripping, with the result that at the end of the run there is an average sample of the gas produced for each furnace under the fireman's charge.

When the fireman finishes his run he takes off the bottles, labels them, putting on his name and date, and the next day the chief engineer analyzes the gas and the fireman is given credit for the carbon dioxide found. The bonuses run from 15 cents a day to 70 cents. If the fireman shows an average percentage of five, or less, he is taken off the job, because he would be an expensive man if he worked for nothing.

If the analysis shows six per cent. of CO_2, he is paid 15 cents a day extra, and so on up to 70 cents, according to the following table:

Per Cent.	Cents.
6	15
6½	20
7	25
7½	30
8	35
8½	40
9	45
10	50
11	55
12	60
13	65
14	70

It has been found impractical to try to go beyond 14 per cent. because losses in other directions more than offset the increased gain through higher CO_2. Experience shows that with this kind of an incentive the fireman will pay a great deal better attention to his business, and instead of firing a few scoops of coal into the middle of the grates he will fire often, and fire thin, keeping his grates thoroughly covered.

The consumption of coal at the plant where this plan has been enforced is from 12 to 16 tons a day, and the saving, after paying the bonuses of the fireman, has averaged from $6 to $8 a day. The institution of this system has proved to be the most profitable investment made at this plant during the past year.

In a discussion of the steam-turbine centrifugal pump, C. H. Hurd brought out the conditions under which this type of pump would prove the best investment. His paper read in part as follows:

In considering the economical advantages of the various types of engines or pumps, it is necessary to take into account not only the first cost or interest upon invested capital, but also the depreciation and maintenance as well as the operating expenses. Unless all of these items are considered it is not possible to come to a proper conclusion as to which would be the most economical type.

In comparing the reciprocating and turbine type of pumps it is also necessary to consider not only the cost of the units themselves, but also the cost of foundations, buildings and boiler plant, as it will readily be seen that the total cost of delivering water will include both the fixed charges and operating expense of the entire plant. It is obvious that a high fixed charge may mean a low cost of operation, or, on the other hand, a smaller investment in pumping equipment

and a correspondingly lower duty may mean a higher boiler-room expense and additional boiler capacity.

In making the comparisons, a plant has been selected with an average pumpage of 10,000,000 gallons for 24 hours, which is perhaps slightly larger than the average water-works plant in the United States. In designing an equipment to take care of the service for the average city it would be considered poor judgment to make the pumping capacity less than twice the average consumption, while perhaps it would be better to divide the installation into three or four units, particularly if the plant is to operate against a direct-pressure system. For the purpose of this paper we assume that the city has a reservoir of such capacity that two 10,000,000-gallon pumps would be sufficient.

As to the cost of such a plant, it is understood that the figures can only be approximate, and that they will vary in both cases according to the conditions and demands. In assuming the operating conditions, we will consider the price of coal at $2.50 per ton, fired into the boilers, and the head pumped against to be 200 feet. The cost of such a pumping plant, using vertical triple-expansion pumping engines, for twenty million total capacity would be about $180,000. With interest and depreciation at 9 per cent., the fixed charge would amount to $16,200 per year, or $4.33 per million gallons pumped. It would be fair to assume that the centrifugal pumping plant, to do like service, including boilers as above, would not cost more than $90,000, making a fixed charge per million gallons of $2.16.

In the case of the triple-expansion engines under most favorable conditions, the station duty would not exceed one hundred and fifty million foot-pounds per one thousand pounds of steam. With the centrifugals it would be fair to assume a duty of eighty million foot-pounds per one thousand pounds of steam; in both cases the water evaporated to be six pounds per pound of coal. One hundred and fifty million foot-pounds duty is equivalent to 13.2 pounds of steam per water horsepower-hour; eighty million duty is equivalent to 24.75 pounds per water horsepower-hour. By using 13.2 pounds per water horsepower-hour and a total head of 200 feet, the steam required is 11,232 pounds per million gallons pumped, and with 6 pounds evaporation, 1872 pounds of coal will be used, which, at $2.50 a ton, would mean that the cost of fuel is $2.34 per million gallons pumped. Also for the equipment using 24.75 pounds per water horsepower-hour, 20,800 pounds of steam are required per million gallons pumped, and with the same evaporation, 3460 pounds of coal would be used at a cost of $4.33 per million gallons.

Assuming labor, depreciation and maintenance to be the same in both cases, this gives a difference of $8100 per year in the fixed charges, in favor of the centrifugal plant, and approximately $7300 per year difference in operation, in favor of the triple-expansion installation, or a net balance of about $800 per year in favor of the less expensive plant. With an average pumpage of ten million gallons per day, this would represent a saving of $0.21 per million gallons pumped.

These figures are not intended to be accurate in all the various details, but are fairly approximate for the conditions assumed. The writer, in making estimates on cost of plants of this character, has come to the following conclusions: For large water-works plants, where the pumps will be in almost constant service, the high-duty pumping engine is the better investment. This is particularly true in districts having a high cost of fuel. On the other hand, for smaller plants and plants where the machinery is liable to be out of service more than half the time, and for auxiliary equipment, the turbine-driven centrifugal will show a marked economy over the larger and more expensive unit.

Election of officers resulted as follows: Alba L. Holmes, Grand Rapids, Mich., president; William J. Scroggins, Wheeling, W. Va., vice-president; William Allen Veach, Newark, O., secretary, and A. W. Irman, Massillon, O., treasurer.

Denver Convention of Steam Engineers

The convention of the International Union of Steam Engineers, held at Denver, Colo., during the week beginning September 12, was a success in every way. During the several business sessions of the convention many matters of importance conducive to the welfare of the organization were disposed of. The many features of entertainment which had been arranged by a competent committee were thoroughly enjoyed by the delegates and guests.

The election of officers resulted as follows: Matthew Comerford, president, St. Paul, Minn.; M. Snellings, first vice-president, Washington, D. C.; T. J. Roberts, second vice-president, Oakland, Cal.; Michael Kelley, third vice-president, Cornwall-on-the-Hudson, N. Y.; Michael Murphy, fourth vice-president, New York City, N. Y.; P. J. Finley, fifth vice-president, Pittsburg, Penn.; James J. Hannahan, secretary-treasurer, Chicago, Ill.; S. J. Stevenson, chairman of board of trustees, Chicago, Ill.

It was decided to hold the next convention at St. Paul, Minn., in September, 1911.

Belgian Exposition

In the August 30 number we published an article on the Belgian Exposition at Brussels, and in the last paragraph gave credence to a report prevalent at the time that the exposition had been practically destroyed by fire. We have since found that the early reports were greatly exaggerated. Only the Belgian section of food products, etc., and a part of the English section, all told less than a tenth part of the exposition, were destroyed. The conflagration caused but a few days' inconvenience and did not seriously interfere with the exposition.

OBITUARY

Frederick M. Wheeler, inventor of the Wheeler surface condenser, died at his summer residence at Westhampton, L. I., on September 16, of heart disease. He was born in Brooklyn in 1848 and was a descendant of an old New England family.

Mr. Wheeler was educated at the Summit Academy and the Brooklyn Polytechnic Institute and later studied mechanical engineering for four years under Henry J. Davison, of New York City, one of the leading mechanical engineers of that time. He afterward took up hydraulic and marine engineering as a specialty, and for over thirty-four years was associated with the George F. Blake Manufacturing Company. He was a director and secretary of the company, and later a director of the International Steam Pump Company, which corporation absorbed the Blake company, the Worthington and other hydraulic works. Mr. Wheeler organized the Wheeler Condenser and Engineering Company, whose works are located at Carteret, N. J., and was also officially connected with the Ludlow Valve Manufacturing Company. He was one of the charter members of the American Society of Mechanical Engineers and the Society of Naval Architects and Marine Engineers, besides being a member of the American Society of Naval Engineers and the Engineers' Club. He served as a member of the advisory council of the Engineering Congress at the Columbian exposition.

Mr. Wheeler was prominent in social circles in Montclair, his permanent home, and was the founder of the Montclair Equestrian Club, which subsequently became the Montclair and Essex County Country Club. He was also one of the founders of the Outlook Club, one of the early members of the Montclair Club, and for many years one of its board of directors. While assisting in the various public improvements from time to time, the subject which interested him most was that of public parks for Montclair, which subject he agitated for many years until it became a reality.

He is survived by a widow, two daughters and two sons.

NEW INVENTIONS

Printed copies of patents are furnished by the Patent Office at 5c. each. Address the Commissioner of Patents, Washington, D. C.

PRIME MOVERS

ELASTIC FLUID TURBINE. William James Albert London, Manchester, England, assignor to the Westinghouse Machine Company, a Corporation of Pennsylvania. 969,891.

WAVE MOTOR. William L. Raht, San Diego, Cal. 969,903.

HYDRAULIC POWER SYSTEM. Thomas A. Macdonald, Clifton, N. J., assignor of sixteen and one-third one-hundredths to Isaiah E. Zimmerman, Paterson, N. J., sixteen and one-third one-hundredths to William H. Castles, Kingsland, N. J., and sixteen and one-third one-hundredths to George Lentz, North Arlington, N. J. 969,967.

TURBINE. Paul F. Nydegger, Elizabeth, N. J., assignor to the Singer Manufacturing Company, a Corporation of New Jersey. 969,974.

WAVE MOTOR. Nathaniel O. Harmon, Los Angeles, Cal. 970,048.

INTERNAL COMBUSTION ENGINE. Simon Lake, Bridgeport, Conn. 970,063.

STEAM TURBINE. Herman Peters, Hamburg, Germany. 970,106.

TURBINE. Willis G. Dodd, of San Francisco, Cal. 970,193.

GAS TURBINE. Erik August Forsberg and Birger Ljungstrom, Stockholm, Sweden. 970,204.

INTERNAL COMBUSTION ENGINE. Leon Auguste Martha, Paris, France. 970,251.

FLUID ACTUATED MOTOR. William Snee, West Elizabeth, Penn. assignor to American Power and Manufacturing Company, Pittsburg, Penn., a Corporation of Delaware. 970,404.

BOILERS, FURNACES AND GAS PRODUCERS

LIQUID FUEL BURNER. Bradford R. Phillips, Cement, Tex. 969,978.

GAS PRODUCER. William B. Chapman, New York, N. Y., assignor to Walter B. Hopping, Trustee, New York, N. Y. 970,017.

OIL BURNER. Mollie J. Wright and William L. Wright, Dunlap, Kan. 970,156.

OIL BURNER. William H. Bradley, Half Moon Bay, Cal. 970,300.

LIQUID FUEL BURNER. Jacob P. Hollenbeck, Frederick, Okla., assignor of one-half to John W. Fondren, Frederick, Okla. 970,335.

OIL OR GAS BURNER. Antone W. Andorfer, Shinglehouse, Penn. 970,363.

GAS BURNER. Ferdinand Mummelthey, Coraopolis, Penn., assignor to Graham Nut Company, Pittsburg, Penn., a Corporation of Pennsylvania. 970,382.

FURNACE. Maurice Blanchard, Wardner, Idaho. 970,417.

POWER PLANT AUXILIARIES AND APPLIANCES

GAGE COCK. John Rowland Brown, Mansfield, O., assignor to Ohio Brass Company, Mansfield, O., a Corporation of New Jersey. 969,770.

STOP VALVE. Thomas Pollock, Folkestone, England, assignor to the Dover Engineering Works, Limited, Dover, England. 969,803.

GAS PRODUCER REGULATING VALVE. Henry I. Lea, Pittsburg, Penn., assignor to the Westinghouse Machine Company, a Corporation of Pennsylvania. 969,884.

PRIMING CHECK VALVE. Joseph A. H. Provost, Van Anda, Texada Island, British Columbia, Canada. 969,902,

CARBURETER. Frederick John Cox, Kentish Town, London, England. 969,941.

CARBURETING APPARATUS. Frederick Ostermayer, Elizabeth, N. J. 970,102.

VALVE SPRING REMOVER. John Turner, Colorado Springs, Colo. 970,141.

FLOAT CONTROLLED VALVE. Winfield S. Bellows, Steelton, Penn. 970,167.

AUTOMATIC FUEL FEED MECHANISM FOR FURNACES. George Dinkel, Jersey City, N. J. 970,191.

METHOD OF COVERING PIPES. Eli A. Kellam, New York, N. Y. 970,235.

STOP AND THROUGH-WAY VALVE. Frederic William Leevers, Walthamstow, England. 970,245.

PUMP VALVE. George E. Mack, Detroit, Mich. 970,249.

FLUID PRESSURE VALVE REGULATOR. Thomas M. Wilkins, East Randolph, N. Y., assignor of five-sixteenths to W. E. Zierden, Johnsonburg, Penn., one-fourth to John G. Whitmore, Ridgway, Penn., and seven-sixteenths to Ziermore Regulator Company, Johnsonburg, Penn., a Corporation of Pennsylvania. 970,284.

LUBRICATING VALVE. Thomas Julian Barbre. Denver, Colo. 970,292.

FURNACE DOOR. George O'Neill, Moncton, New Brunswick, Canada. 970,347.

RELIEF OR SAFETY VALVE. William L. Osborne, Chicago, Ill. 970,348.

VALVE. Matthew H. Owen, Chicago, Ill. 970,387.

WATER LEVEL REGULATOR. Noiberto Guy Copley, Fostoria, O., assignor to the "S. C." Regulator Company, Fostoria, O., a Corporation of Ohio. 970,421.

CARBURETER FOR INTERNAL COMBUSTION ENGINES. Fitzwilliam Richard Davis, Kew Gardens, England. 970,429.

AUTOMATIC FUEL FEED MECHANISM FOR FURNACES. George Dinkel, Jersey City, N. J. 970,432.

PUMP. George J. Dourte, Longmont, Colo. 970,435.

OIL FILTER. John Francis Deems, New York, N. Y. 966,173.

WATER-GAGE GLASS. Nicholas Zucke, Sparks, Nev. 966,389.

REGULATING VALVE FOR DASHPOTS. Christian Dorn, Philadelphia, Penn., assignor to James F. Marshall, Philadelphia, Penn. 966,320.

WATER-GAGE ATTACHMENT. Zachariah C. Ferris, Caldor, Cal. 966,323.

AUTOMATIC FEED REGULATING AND MIXING DEVICE FOR CRUDE-OIL BURNERS. Joseph B. Willings, Maricopa, Cal. 966,373.

CARBURETER. Evert P. Brooks, Cincinnati, Ohio, assignor by direct and mesne assignments, of one-third to Charles A. Gibson, and one-third to John Remmers, Jr., Cincinnati, Ohio. 966,381.

VALVE. Robert W. Elder, Finch, W. Va. 966,390.

ELECTRICAL INVENTIONS AND APPLICATIONS

ALTERNATING CURRENT MAGNET. Arthur Simon, Milwaukee, Wis., assignor to the Cutler-Hammer Manufacturing Company, Milwaukee, Wis., a Corporation of Wisconsin. 969,890.

PORTABLE ELECTRIC HEATER. Frank L. Dyer, Montclair, N. J. 969,849.

ELECTROLYTIC APPARATUS. William Thum, Hammond, Ind. 969,921.

PROCESS OF ELECTRICAL SEPARATION. Henry Azor Wentworth, Lynn, Mass., assignor to Huff Electrostatic Separator Company, Boston, Mass., a Corporation of Maine. 970,001.

ELECTRIC CUT OUT. Thomas E. Murray, New York, N. Y. 970,088.

ELECTRICAL PROTECTIVE APPARATUS. Jules A. Birsfield, Rochester, N. Y., assignor to Charles A. Brown and Lynn A. Williams, Chicago, Ill., a Copartnership. 970,169.

ELECTROMAGNETIC DEVICE. William S. Burnett, Milwaukee, Wis., assignor to Morse Code Signal Company, Milwaukee, Wis., a Corporation of Wisconsin. 970,180.

PROTECTED OR ARMOR CLAD ELECTRIC SWITCH FUSE. Herbert Henry Berry and William J. Markham, London, England. 970,296.

STORAGE BATTERY. Rufus N. Chamberlain, Depew, N. Y., assignor to Gould Storage Battery Company, New York, N. Y. 970,420.

POWER-PLANT TOOLS

CUTTER FOR PIPE AND THE LIKE. John R. Hamilton, Yonkers, N. Y., assignor to Ventura Alarm Company, Dover, N. J., a Corporation of New Jersey. 969,858.

WRENCH. Alix Fresko and Bela W. Fresko, Amsterdam, O. 970,209.

RATCHET WRENCH. Ambus M. Fulkerson, Sac and Fox Agency, Okla. 970,320.

WRENCH. Uriah C. Greggs, Terre Haute, Ind. 970,371.

DRILL. Charles Seals, Lovilla, Iowa, assignor of one-half to William Livix, Lovilla, Iowa. 970,401.

ENGINEERING SOCIETIES

AMERICAN SOCIETY OF MECHANICAL ENGINEERS

Pres., George Westinghouse; sec., Calvin W. Rice, Engineering Societies building, 29 West 39th St., New York. Monthly meetings in New York City.

NATIONAL ELECTRIC LIGHT ASSOCIATION

Pres., W. W. Freeman, Brooklyn, N. Y.; sec., T. C. Martin, 31 West Thirty-ninth St., New York.

AMERICAN SOCIETY OF NAVAL ENGINEERS

Pres., Engineer-in-Chief Hutch I. Cone, U .S. N.; sec. and treas., Lieutenant Henry C. Dinger, U. S. N., Bureau of Steam Engineering, Navy Department, Washington, D. C.

AMERICAN BOILER MANUFACTURERS' ASSOCIATION

Pres., E. D. Meier, 11 Broadway, New York; sec., J. D. Farasey, cor. 37th St. and Erie Railroad, Cleveland, O. Next annual meeting at Chicago, Oct. 10-13, 1910.

WESTERN SOCIETY OF ENGINEERS

Pres., J. W. Alvord; sec., J. H. Warder, 1735 Monadnock Block, Chicago, Ill.

ENGINEERS' SOCIETY OF WESTERN PENNSYLVANIA

Pres., E. K. Morse; sec., E. K. Hiles, Oliver building, Pittsburg, Penn. Meetings 1st and 3d Tuesdays.

AMERICAN INSTITUTE OF ELECTRICAL ENGINEERS

Pres., Dugald C. Jackson; sec., Ralph W. Pope, 33 W. Thirty-ninth St., New York. Meetings monthly.

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

Pres., Prof. J. D. Hoffman; sec., William M. Mackay, P. O. Box 1818, New York City.

NATIONAL ASSOCIATION OF STATIONARY ENGINEERS

Pres., Carl S. Pearse, Denver, Colo.; sec., F. W. Raven, 325 Dearborn street, Chicago, Ill. Next convention, Cincinnati, Ohio.

UNIVERSAL CRAFTSMEN COUNCIL OF ENGINEERS

Grand Worthy Chief, John Cope; sec., J. U. Bunce, Hotel Statler, Buffalo, N. Y. Next annual meeting in Philadelphia, Penn., week commencing Monday, August 7, 1911.

AMERICAN ORDER OF STEAM ENGINEERS

Supr. Chief Engr., Frederick Markoe, Philadelphia, Pa.; Supr. Cor. Engr., William S. Wetzler, 753 N. Forty-fourth St., Philadelphia, Pa. Next meeting at Philadelphia, June, 1911.

NATIONAL MARINE ENGINEERS BENEFICIAL ASSOCIATIONS.

Pres., William F. Yates, New York, N. Y.; sec., George A. Grubb, 1040 Dakin street, Chicago, Ill. Next meeting, St. Louis, Mo., January 16-21, 1911.

OHIO SOCIETY OF MECHANICAL ELECTRICAL AND STEAM ENGINEERS

Pres., O. F. Rabbe; sec. and treas., Prof. F. E. Sanborn, Ohio State University, Columbus, Ohio.

INTERNATIONAL MASTER BOILER MAKERS' ASSOCIATION

Pres., A. N. Lucas; sec., Harry D. Vaught, 95 Liberty street, New York. Next meeting at Omaha, Neb., May, 1911.

INTERNATIONAL UNION OF STEAM ENGINEERS

Pres., Matt. Comerford; sec., J. G. Hannahan, Chicago, Ill. Next meeting at St. Paul, Minn., September, 1911.

NATIONAL DISTRICT HEATING ASSOCIATION.

Pres., G. W. Wright, Baltimore, Md.; sec. and treas., D. L. Gaskill, Greenville, O.